GEORGIA INSTITUTE OF TECHNOLOGY OFFICE OF CONTRACT ADMINISTRATION

PROJECT ADMINISTRATION DATA SHEET

[x] ORIGINAL [ ] REVISION NO. ______

Project No. A-3345 (subproject is E-25-664) GTRI/~~GIT~~x DATE 10/28/82

Project Director: Walter Hendrix ~~School~~/Lab TAL

Sponsor: HQ Ballistic Missile Office, AFSC, Norton AFB, CA 92409

Type Agreement: Contract #F33657-82-G-2083-R901*

Award Period: From 9/1/82 To ~~5/31/83~~ 9/30/83 (Performance) 7/31/83 (Reports)

Sponsor Amount: Total Estimated: $ 89,953 Funded: $ 89,953 (includes $20,612 on sub-project E-26-664)

Cost Sharing Amount: $ ______ Cost Sharing No: ______

Title: The M-X Deep Basing Heat Pipes for Thermal Dissipation

ADMINISTRATIVE DATA OCA Contact John W. Burdette x4820

1) Sponsor Technical Contact:

Capt. C. Meyer (SYB)
HQ Ballistic Missile Office
AFSC
Norton AFB, CA 92409
(714) 382-3263

2) Sponsor Admin/Contractual Matters:

Mr. Thomas A. Bryant
ONR RR
206 O'Keefe Bldg.
Georgia Institute of Technology
Atlanta, GA 30332

Defense Priority Rating: DO-C9

Military Security Classification: ______
(or) Company/Industrial Proprietary: ______

RESTRICTIONS

See Attached Gov't Supplemental Information Sheet for Additional Requirements.

Travel: Foreign travel must have prior approval -- Contact OCA in each case. Domestic travel requires sponsor approval where total will exceed greater of $500 or 125% of approved proposal budget category.

Equipment: Title vests with Gov't; except that items costing $1,000 or less vest with GIT providing prior written approval to purchase received from Contracting Officer.

COMMENTS:

*Basic Order Agreement No. F33657-82-G-2083 on file with Lab Director's Office/TAL

NOTE: Mod 1 is incorporated in this initiation changing the paying station code appearing in Block 13 of Order R901 from S09206 to S1102A

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GEORGIA INSTITUTE OF TECHNOLOGY OFFICE OF CONTRACT ADMINISTRATION

SPONSORED PROJECT TERMINATION/CLOSEOUT SHEET

Date 12/18/84

Project No. A-3345 ~~XXXXX~~ School/Lab TAL

Includes Subproject No.(s) E-25-664/G. T. Colwell

Project Director(s) Walter Hendrix GTRI / ~~XXXX~~

Sponsor H Q Ballistic Missile Office, AFSC, Norton AFB, GA 92409

Title The M-X Deep Basing Heat Pipes for Thermal Dissipation

Effective Completion Date: 9/30/83 (Performance) 9/30/83 (Reports)

Grant/Contract Closeout Actions Remaining:

- [ ] None
- [X] Final Invoice or Final Fiscal Report
- [X] Closing Documents
- [X] Final Report of Inventions
- [X] Govt. Property Inventory & Related Certificate
- [ ] Classified Material Certificate
- [ ] Other ______

Continues Project No. ______ Continued by Project No. ______

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Other M. Heyser; A. Jones

H-3518

ENGINEERING EXPERIMENT STATION
Georgia Institute of Technology
A Unit of the University System of Georgia
Atlanta, Georgia 30332

October 15, 1982

Captain Donald Carroll
BMO/SYBU
Norton Air Force Base, California 92409

Subject: M-X Deep Basing Heat Pipes for Thermal Dissipation Monthly Progress/Status Meeting Report

Dear Captain Carroll:

This report summarizes activities on Research Project A3345 (F33657-82-G-2083, R901) for the period 01 September 82 through 30 September 82. Planned activities for the period 01 October 82 through 31 October 82 are also presented.

1. Activities during September, 1982 centered primarily around investigations of the heat transfer characteristics of the rock environment and preliminary heat pipe design. The applicable governing differential equation (GDE) for the problem at hand involves the conduction of heat to a region bounded internally by an infinite circular cylinder. This problem has been treated in a number of mathematical and engineering texts giving rise to both general and closed-form solutions for the cases of constant cylinder surface temperature and constant cylinder heat flux. It is expected that the constant heat flux case will be of most importance for analyzing this problem.

Since the general solution for the constant heat flux case is hard to evaluate, being left in the form of an infinite integral of an algebraic Bessel Function expression, efforts will first center around evaluation of the use of the closed-form solutions for determining temperature profiles in the rock at various times. These analytical solutions include a small time solution, a large time solution, and a solution obtained by idealizing the problem as that of a continuous line source in an infinite medium.

Calculations of temperature profiles at various times will be made for each solution using representative rock properties and expected heat flux rates. The temperature profiles obtained will be compared against each other and against profiles obtained by numerical solution of the GDE using a finite difference method. The applicability of the closed form solutions will also be checked against a limited amount of data found in the literature which was obtained from the general solution for one particular case. This data is in dimensionless form which will allow direct comparison of the closed form solutions with the general solution over a limited range of values. Based on all of these calculations, the correctness and range of applicability of each of the closed-form solutions will be determined. Should these closed form solutions prove unsuitable, analysis of the heat transfer characteristics in the rock will rely either on evaluating the general solution or complete numerical solution of the GDE.

Preliminary investigation of heat pipe design has involved investigation of capillary and entrainment limitations, pressure distributions in the vapor phase, and boiling limitations in the capillary structure. Working fluids being looked at presently are methanol and water. The overall thermal conductances of several heat pipe designs are being investigated. It appears that the overriding factor in the thermal coupling of the working fluid to the heat sink will be the heat transfer in the rock leaving a fair degree of latitude, from a thermal standpoint, in selection of heat pipe materials. Consequently, a number of materials which would allow fabrication of a flexible heat pipe are being examined.

2. Planned activities for October, 1982 include continued investigation of the heat transfer characteristics of the rock with respect to establishing the appropriate analytical treatment treatment and delineation of specific data related to thermal loads and rock properties and expected heat pipe applications based on conversations with BMO. Preliminary design of the heat pipes based on the above data will continue. A meeting between Georgia Tech, BMO and other appropriate personnel is planned for the afternoon of 19 October 82 at Norton Air Force Base.

If there are any questions concerning this report or the status of the program, please call Walter Hendrix or Gene Colwell.

Respectfully submitted,

Walter A. Hendrix, P.E.
Co-Principal Investigator
Technology Applications Laboratory

Gene T. Colwell, P.E., PhD
Co-Principal Investigator
School of Mechanical Engineering

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H 2348

ENGINEERING EXPERIMENT STATION
Georgia Institute of Technology
A Unit of the University System of Georgia
Atlanta, Georgia 30332

IN CONFIDENCE

November 19, 1982

Captain Clark Myers
HQ Ballistic Missile Office
AFSC
Norton AFB, California 92409

Subject: M-X Deep Basing Heat Pipes for Thermal Dissipation Monthly Progress/Status Meeting Report for October, 1982

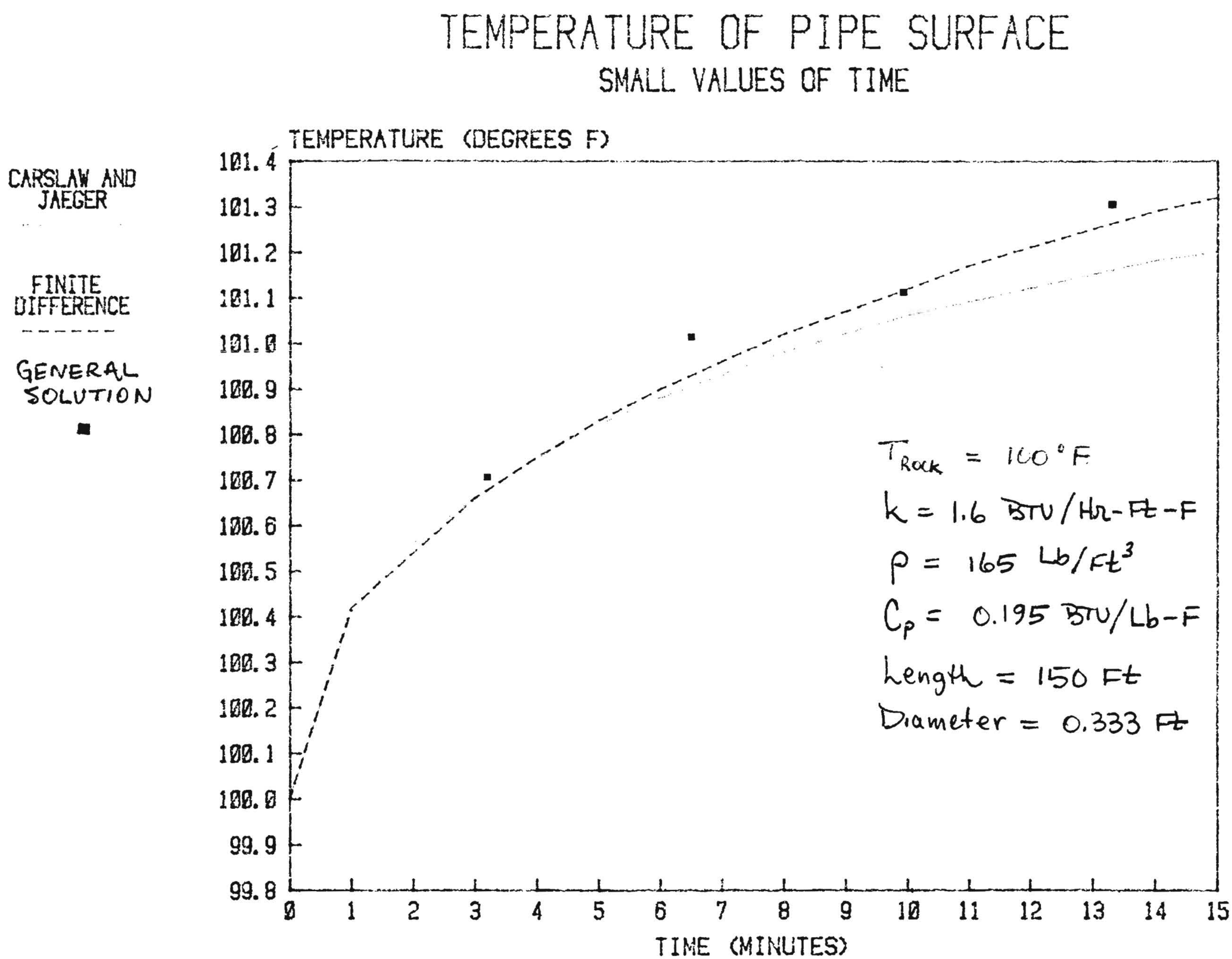
Dear Captain Myers:

This report summarizes activities on Research Project A3345 (F33657-82-G-2083, R901) for the period 01 October 82 through 31 October 82. Planned activities for the period 01 November 82 through 30 November 82 are also presented.

1. Rock Heat Transfer Characteristics. Investigation of the heat transfer characteristics of the rock is virtually complete. Expressions have been found which accurately yield rock temperatures as a function of time and radial distance for a heat pipe transferring heat to the rock at constant flux. The temperature profiles expected to be of most interest are those during the first few minutes after heat pipe start-up and after a year of operation.

   Figure 1 presents pipe surface temperature as a function of time for small values of time. The temperature values were calculated by finite difference and using an algebraic expression developed by Carslaw and Jaeger. These two expressions are seen to agree for the time interval, 0-5 minutes which indicates the Carslaw and Jaeger expression, which is much simpler and quicker to use, will provide suitable accuracy for modelling and parametric studies of heat pipe start-up. Four points calculated from literature data obtained from the general solution of the governing differential equation are also plotted in Figure 1. These limited data agree with both solutions in the 0-5 minute interval and with the finite difference method after 5 minutes indicating it would be suitable for use, if needed, during this timeframe.

   Figure 2 presents rock temperatures as a function of radial distance from the heat pipe surface at time equal to 10,000 hours. The temperature values were calculated by finite difference, using an expression for large values of time developed by Carslaw and Jaeger, and using an expression obtained by idealizing the heat pipe as an infinite line source. The Carslaw and Jaeger solution and the line source solution are seen to agree very well at distances less



TEMPERATURE OF PIPE SURFACE
SMALL VALUES OF TIME
TEMPERATURE (DEGREES F)
CARSLAW AND JAEGER
FINITE DIFFERENCE
GENERAL SOLUTION
101.4
101.3
101.2
101.1
101.0
100.9
100.8
100.7
100.6
100.5
100.4
100.3
100.2
100.1
100.0
99.9
99.8
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15
TIME (MINUTES)
T_{Rock} = 100 °F
k = 1.6 BTU/Hr-Ft-F
ρ = 165 Lb/Ft³
C_p = 0.195 BTU/Lb-F
Length = 150 Ft
Diameter = 0.333 Ft

Figure 1

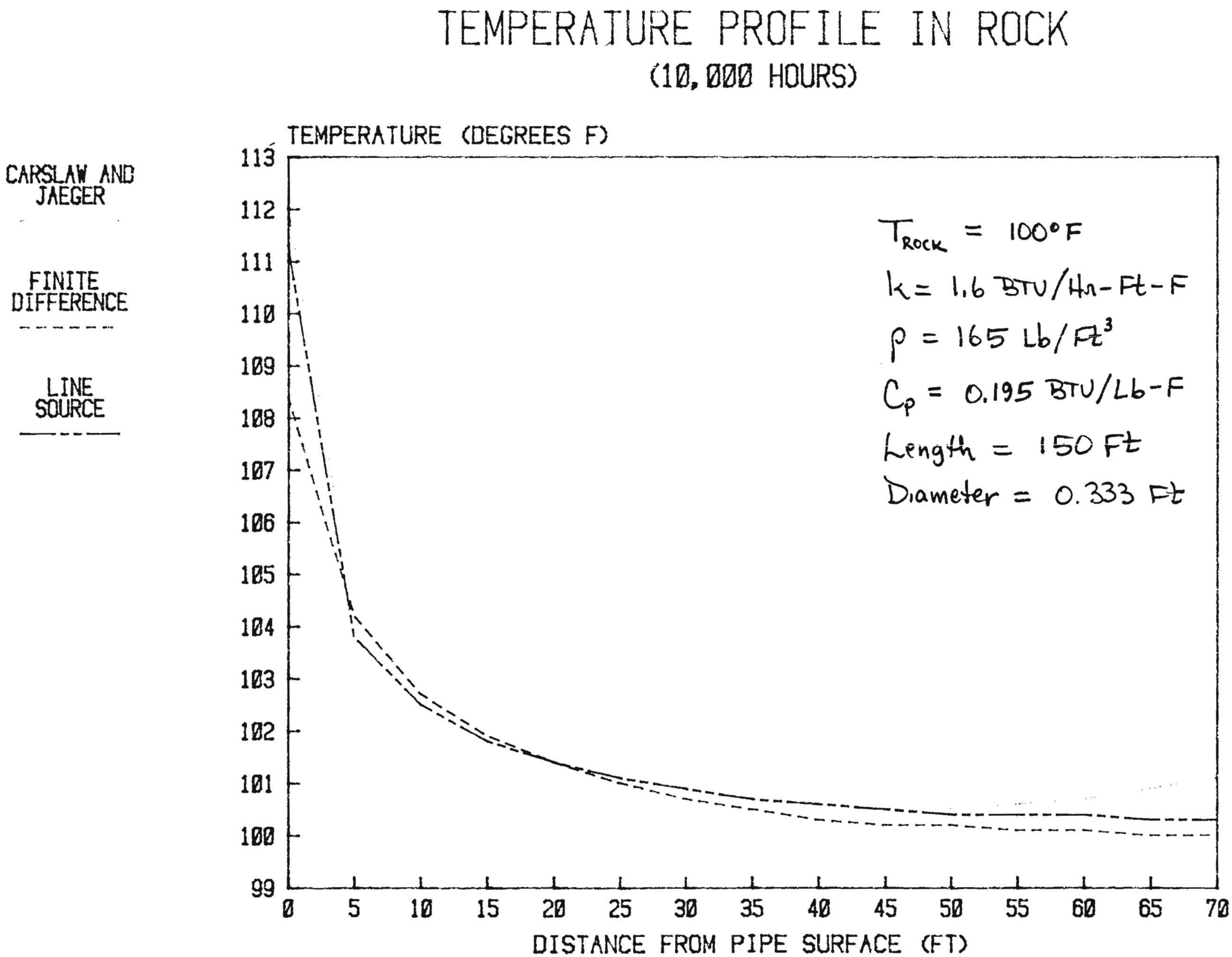

TEMPERATURE PROFILE IN ROCK
(10,000 HOURS)
TEMPERATURE (DEGREES F)
CARSLAW AND JAEGER
FINITE DIFFERENCE
LINE SOURCE
T_{ROCK} = 100°F
k = 1.6 BTU/Hr-Ft-F
ρ = 165 Lb/Ft^3
C_p = 0.195 BTU/Lb-F
Length = 150 Ft
Diameter = 0.333 Ft
113
112
111
110
109
108
107
106
105
104
103
102
101
100
99
0
5
10
15
20
25
30
35
40
45
50
55
60
65
70
DISTANCE FROM PIPE SURFACE (FT)

Figure 2

than 5 feet from the pipe surface, even for distances very close to the pipe surface where the line source solution does not accurately reflect the physical situation. The table below gives actual temperature values calculated, at 10,000 hours for various radial distance from the pipe surface, by the Carslaw and Jaeger solution and using literature data obtained from the general solution of the governing differential equation.

Time = 10,000 Hours

	Radial Distance (ft)			
	0.000	0.166	0.666	1.500
Carslaw and Jaeger	112.0	110.4	108.3	106.8
General	111.9	110.3	108.2	106.7

These calculated values indicate that the Carslaw and Jaeger expression for large time values, which is easy to use, will accurately reflect temperatures at radial distances less than 5 feet. Due to the method used by the computer plotting system which generated all of the graphs in this report (i.e., straight lines drawn between a limited number of points), the curves appear as straight lines below 5 feet radial distance rather than having a more exponential type of shape as indicated by the numbers in the above table. This plotting deficiency may be rectified by using a different plotting routine or by drawing them by hand.

In the range of 5 feet to 20 feet, all three methods agree fairly well, so again, because of its simplicity, the Carslaw and Jaeger expression will be used. There are no general solution data available, at this point in time, for radial distances greater than 1.5 feet.

At distances greater than 20 feet, the finite difference method appears to be the most accurate. The line source expression is a series expansion which can probably be made to agree with finite difference if enough terms are considered. In this case, since it is simpler to use, it would probably be preferred over finite difference for parametric studies.

Figures 3 and 4 give examples of the types of parametric studies which will be performed. Figure 3 shows the rock temperature profile for various heat fluxes. The heat fluxes can be changed by 1) increasing the heat load into the pipe at constant pipe length and diameter, or 2) by using different length pipes of the same diameter with a constant evaporator heat load. Figure 4 shows the rock temperature profile for various diameter pipes all having the same heat flux. This may be

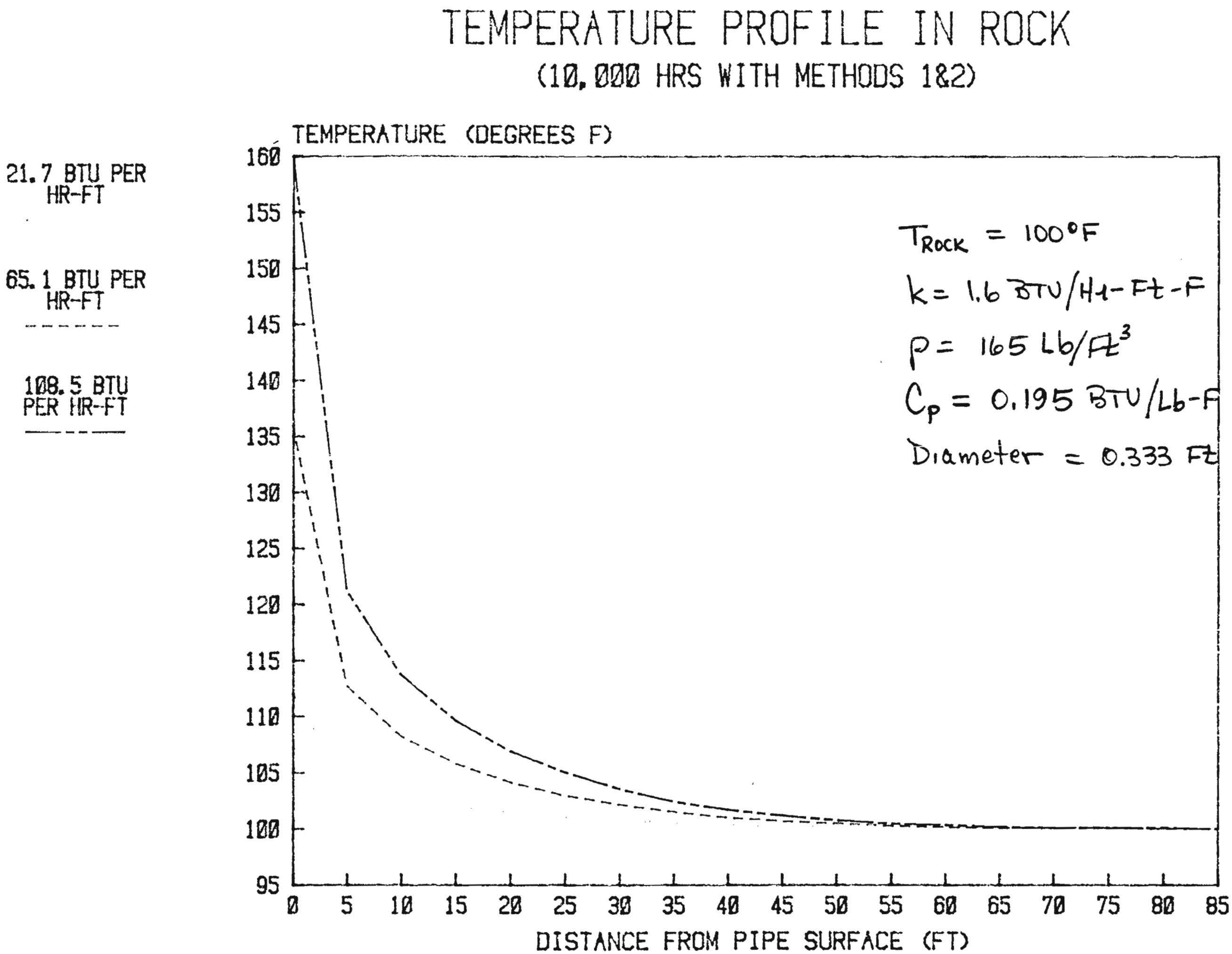
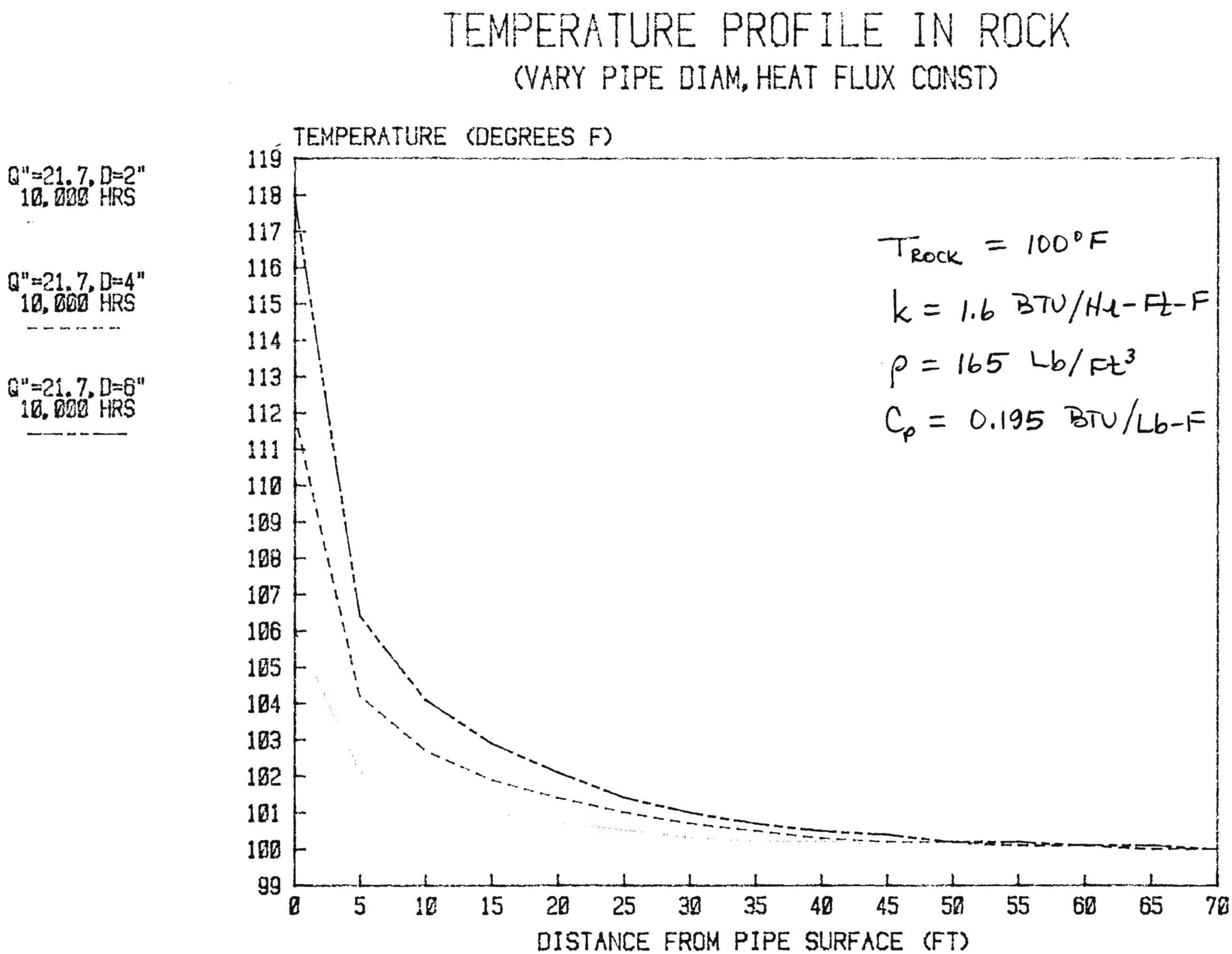

TEMPERATURE PROFILE IN ROCK
(10,000 HRS WITH METHODS 1&2)
TEMPERATURE (DEGREES F)
21.7 BTU PER HR-FT
65.1 BTU PER HR-FT
108.5 BTU PER HR-FT
T_ROCK = 100°F
k = 1.6 BTU/Hr-Ft-F
ρ = 165 Lb/Ft³
Cp = 0.195 BTU/Lb-F
Diameter = 0.333 Ft
160
155
150
145
140
135
130
125
120
115
110
105
100
95
0
5
10
15
20
25
30
35
40
45
50
55
60
65
70
75
80
85
DISTANCE FROM PIPE SURFACE (FT)

Figure 3



TEMPERATURE PROFILE IN ROCK
(VARY PIPE DIAM, HEAT FLUX CONST)
TEMPERATURE (DEGREES F)
Q"=21.7, D=2"
10,000 HRS
Q"=21.7, D=4"
10,000 HRS
Q"=21.7, D=6"
10,000 HRS
T_ROCK = 100°F
k = 1.6 BTU/Hr-Ft-F
ρ = 165 Lb/Ft³
Cp = 0.195 BTU/Lb-F
119
118
117
116
115
114
113
112
111
110
109
108
107
106
105
104
103
102
101
100
99
0
5
10
15
20
25
30
35
40
45
50
55
60
65
70
DISTANCE FROM PIPE SURFACE (FT)

Figure 4

accomplished by increasing evaporator heat load for larger diameter pipes of the same length or by using shorter but larger diameter pipes at constant evaporator heat load. The plots in both of these figures were obtained using Carslaw and Jaegar up to 20 feet from the pipe surface, then finite difference until a rock ambient temperature was obtained.

The rock heat transfer solutions investigated thus far correspond to the simplest, but most conservative, case to analyze, that of one-dimensional radial heat flow to infinite surroundings at uniform initial temperature. Should two-dimensional heat flow, rock environment with some initial temperature profile, coupling between heat pipes, or any combination of the above conditions require analysis, this may be accomplished, though with added complexity, by finite difference methods.

2. Rock Physical Properties. Considerable investigation of the literature has centered around determination of representative rock physical properties to be used in performing analyses. Lack of specific information regarding the expected location of the M-X missile deep bases has resulted in the selection of a range of physical properties to be used. These ranges are given below. More specific values for these properties may be used based on guidance from BMO and continued literature search for this type of information.

Thermal Conductivity, BTU/hr-ft-°F	0.5 - 2.5
Specific Heat at Constant Pressure, BTU/lb-°F	0.15 - 0.25
Density, lb/ft^3	125 - 175

3. Heat Pipe System Design. Investigation of heat pipe design has continued. Much of this activity has been focused at finding materials of construction which will minimize cost while maximizing survivability. It is felt that both of these objectives may be met by identifying materials such as rubbers or plastics which may be cheaply manufactured and will be flexible. Consideration is also being given to fabrication of the pipe with the wick playing an integral role with respect to mechanical performance. Data is being collected on thermal, mechanical, and chemical performance of various flexible materials which will be compared with traditional heat pipe materials.

Investigation of the internal thermal and fluid dynamics of the heat pipe has continued. Design of the evaporator section has been initiated and will consider heat transfer from hot water, steam, air, and refrigerants such as Freon.

We currently anticipate that the heat pipe heat dissipation systems will be designed as modules capable of handling some discrete portion of a specified load. These modules, which would have different designs for different applications such as power plant waste heat removal, space conditioning, etc., could then be grouped together to handle the complete load whether it be waste heat from the power plant condenser, space conditioning for a portion of funnel facility, or some other application.

4. Planned Activities for November. During this time period we will continue heat pipe system design work and characterization of appropriate materials of fabrication with regard to minimizing cost and maximizing survivability. We hope to be able to schedule a Phase I briefing at Norton Air Force Base in mid-December.

If there are any questions concerning this report or the status of the program, please call Walter Hendrix or Gene Colwell.

Respectfully submitted,

Walter A. Hendrix, P.E.
Co-Principal Investigator
Technology Applications Laboratory

Gene T. Colwell, P.E., PhD
Co-Principal Investigator
School of Mechanical Engineering

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Monthly Progress/Status Report, November 1982

Monthly Progress/Status Report, December 1982

A-3345

ENGINEERING EXPERIMENT STATION
Georgia Institute of Technology
A Unit of the University System of Georgia
Atlanta, Georgia 30332

~~IN CONFIDENCE~~

February 10, 1983

Captain Clark Myers
HQ Ballistic Missile Office
AFSC
Norton AFB, California 92409

Subject: M-X Deep Basing Heat Pipes for Thermal Dissipation
Monthly Progress/Status Meeting Report for
January, 1983

Dear Captain Myers:

This report summarizes activities on Research Project A3345 (F33657-82-G-2083, R901) for the period 01 January 1983 through 31 January 1983. Planned activities for the period 01 February 1983 through 28 February 1983 are also presented.

1. Sink Heat Transfer Characteristics. Work continues related to analysis of the rock heat transfer characteristics. This activity is aimed at investigating the range of interaction between the heat pipes for various spacings and configurations, specifically with regard to effect on time versus temperature profiles. Such information is important to establish expected performance for both the heat pipes and the sink. Efforts are currently directed towards developing the algorithms and computer code needed to perform the analyses.

2. Heat Pipe and Waste Heat Removal System Design. Equations are being developed which couple the heat source, heat pipe, and heat sink together. Once formulated these expressions can be reduced to computer code and used to study transient behavior of the entire system as well as effect of different operating conditions, including heat pipe/header interaction, response to attack, and heat pipe performance versus temperature gradient along header.

3. Heat Pipe Fabrication, Installation, and Maintenance. The bulk of the work related to fabrication (method and materials), installation, and maintenance must necessarily await the completion of thermal design and survivability studies. Survivability will be assessed by studying heat pipe mechanical performance while

Captain Clark Myers
February 10, 1983
Page Two

subject to expected forces, for various materials, geometries, and system configurations, and subsequently determining probability of failure. Once thermal design and survivability studies provide more definition, study of fabrication, installation, and maintenance will be initiated. Presently, data is being collected related to these considerations, including drilling techniques, properties of potential heat pipe materials, and properties of materials which might prove suitable as "thermal grout".

4. Planned Activities for February. The activities described above will be continued during February. In addition, planning and preparation will be performed for a Technology Review Meeting to be held at Norton AFB on 23-24 February 1983.

If there are any questions concerning this report or the status of the program, please call Walter Hendrix or Gene Colwell.

Respectfully submitted,

Walter A. Hendrix, P.E.
Co-Principal Investigator
Technology Applications Laboratory

Gene T. Colwell, P.E., PhD
Co-Principal Investigator
School of Mechanical Engineering

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ENGINEERING EXPERIMENT STATION
Georgia Institute of Technology
A Unit of the University System of Georgia
Atlanta, Georgia 30332

IN CONFIDENCE

March 23, 1983

Captain Clark Myers
HQ Ballistic Missile Office
AFSC
Norton AFB, California 92409

Subject: M-X Deep Basing Heat Pipes for Thermal Dissipation
Monthly Progress/Status Meeting Report for
February, 1983

Dear Captain Myers:

This report summarizes activities on Research Project A-3345 (F33657-82-G-2083-R901) for the period 01 February 1983 through 28 February 1983. Planned activities for the period 01 March 1983 through 31 March 1983 are also presented.

1. Sink Heat Transfer Characteristics. Analysis continues related to modelling of heat pipe interaction. Worst and average case will be investigated first for the staggered top/bottom centerline configuration presented in Phase I. Efforts are presently directed towards writing and debugging the computer code need to study the interaction problem.

2. Heat Pipe and Waste Heat Removal Design. Steady state heat pipe equations which take into account fluid overfill have been developed and are being coded for computer analyses. Work continues directed towards development of transient heat pipe equations, coupling equations for the source and sink, and equations which will describe gravity flow in the heat pipe.

3. Heat Pipe Fabrication, Installation, and Maintenance. Analysis of heat pipe mechanical performance while subject to expected forces has commenced. Data and information collection related to drilling techniques and thermal grouting materials is continuing.

4. Planned Activities for March. Continue work as described above. Make corrections and prepare final drawings for Phase I report.

If there are any questions concerning this report or the status of the program, please call Walter Hendrix or Gene Colwell.

Respectfully submitted,

Walter A. Hendrix, P.E.
Co-Principal Investigator
Technology Applications Laboratory

Gene T. Colwell, P.E., PhD
Co-Principal Investigator
School of Mechanical Engineering

mro

1. Sink Heat Transfer Characteristics

4. Planned Activities for April

Continue work as described above.

If there are any questions concerning this report or the status of the program, please call Walter Hendrix or Gene Colwell.

Respectfully submitted:

Walter A. Hendrix, P.E.
Co-Principal Investigator
Technology Applications Laboratory

Gene T. Colwell, P.E., PhD
Co-Principal Investigator
School of Mechanical Engineering

mro

TAL Advises On Deep Basing

"Deep base" concepts have been studied for a variety of defense applications for a number of years. In fact, the Air Force currently is looking at deep basing as a possible long-term solution of the problem of missile survivability and endurance — for the 1990's and beyond. This would involve tunneling thousands of feet into the earth. Such a facility would be required to operate in the pre-attack mode for at least ten years, and for up to one year independent of external support after post-attack "button-up." The complex would include a tunneling machine that could dig out of the cavern in preparation for missile launch, as well as launch control and life support systems for the crew.

A major problem with this concept is how to get rid of the tremendous amount of waste heat that would be generated by the necessary power plant, equipment and people. The tunnel would be insulated from the surrounding rock environment, which at these depths has normal temperatures ranging from 70° to 100°F.

Enter Georgia Tech. The Technology Applications Lab (TAL) and the School of Mechanical Engineering (ME) submitted a joint proposal to study the feasibility of using heat pipes to dissipate the waste heat to the surrounding rock. The Air Force Ballistic Missile Office funded the study for a nine-month span ending this May.

Heat pipes are passive devices that ideally will act isothermally to conduct waste heat from the source to a "heat sink," where the heat is dumped. They are sealed, fluid-filled tubes with wicks. Waste heat applied at the lower end of the tube causes the liquid in the pipe to evaporate. The vapor rises to the other end of the pipe, where it condenses, and the heat is dissipated to the cooler rock surrounding the pipe. The condensate (fluid) travels back down the wick, and the process begins again.

ME Professor Gene Colwell, a recognized expert with 20 years of experience with heat pipes, is co-principal investigator on the project, along with Walter (Bo) Hendrix, chief of TAL's Process Technology Division. Working with them are Wesley Pidgeon and Michael Brown of TAL and Julio Santander, an ME graduate student.

They have completed a preliminary analysis of the thermal characteristics of the rock environment and are currently involved in the conceptual design phase. "We are executing a conceptual design of a heat pipe heat dissipation system for each practical Deep Base application," said Hendrix. "We also will recommend laboratory and prototype system test programs leading to design information for the manufacture, installation, operation and maintenance of full-scale systems."

Large ECM Upgrade Study Under Way

The Systems Engineering Lab (SEL) is conducting a $3.2-million program to support upgrade of an electronic countermeasures (ECM) system for tactical aircraft. The basic objective of the study is to define requirements for updating a 10-year-old Air Force ECM pod so that it will continue to be an effective ECM asset for the near- and longer-term.

"The ALQ-131 System Update Missionization Study (SUMS), which includes a major subcontract to ARINC Research Corporation, is probably the largest analytical study in EES history to be focused on a single production system," said Project Director Jerry Heckman. "The monthly workload will peak this summer at a level equivalent to about 33 full-time professionals — 23 at EES and 10 at ARINC."

SEL's Countermeasures Development Division has overall responsibility for the project, with major assistance from the Concepts Analysis Division. Electronics and Computer Systems Lab personnel are assisting in the area of antennas. EES work focuses on ways to improve the pod's ECM performance, while ARINC is assessing improvements in logistics support and operational availability.

The SUMS team is hard at work looking at proposed modifications and available technology, and will make recommendations for integrating modifications into the production system. EES task leaders are Larry Stroud, Bud Sears, Steve Livesay, and Vic Tripp. The program sponsor is the Aeronautical Systems Division at Wright-Patterson Air Force Base.

RAIL To Open Eskimo Camp

Nick Currie, Jerome Callahan and Chris Lott of the Radar and Instrumentation Lab (RAIL) are suiting up for a chilly spring. They will be spending the month of May camping out in a double-wide trailer in a remote location above the Arctic Circle!

Their mission? To measure the radar reflectivity of sea ice in order to discriminate between multiyear (thick) and first-year (thin) ice. They will be making low-angle measurements at 10 GHz, 16 GHz and 35 GHz under contract with the Canadian Department of Fisheries and Oceans.

"The ultimate application," said Currie, "is to allow the petroleum industry to bring icebreaker tankers in so that drilling can be conducted year-round. Currently, they have to shut down about seven months a year. With a shipboard method of discriminating between safe, thin ice and ship-damaging, thick ice, the icebreakers could open up the Arctic oil fields for 12-month operation."

The intrepid EES trio, plus five Canadians, will set up their radar camp on the northwest side of Baffin Island, just south of Lancaster Sound. Currie says the site is 200 miles east of the north magnetic pole and 100 miles west of Greenland. Besides their long underwear, they will be taking radar equipment and a computer for on-site digital analysis.

Currie says they're looking forward to the entertainment provided by a nearby Eskimo village, as well as observing the local wildlife — mainly seals and polar bears. "We'll have an armed Eskimo guard at all times," he said, "since I understand that polar bears are among the few animals that hunt humans for pleasure."

How did this unusual project come about? A couple of years ago, Currie presented a paper on millimeter waves at the URSI Specialist Conference on Land and Sea Backscatter. There he met a Canadian oil company engineer who was seeking information on solutions to the industry's problems with sea ice. A seed was planted which eventually led to this contract with the Canadian government.

Station News

Georgia Tech Engineering Experiment Station

Volume 13 Number 8 April 1983

EES Holds "Best Ever" External Advisors' Meeting

Members of the EES External Advisory Board took their annual look at the Station April 6-7 and liked what they saw, complimenting EES on the progress made since last year in cooperative effort and planning for the future.

"We are impressed with the quality of your people, your programs, and the fact that you ask us how to improve," said Dr. Joseph A. Saloom, senior vice president and director, Components Technology Center, M/A-Com, Inc. "We also are impressed by what you know about your markets and R&D trends."

The advisors saw "terrific improvement in cooperative effort," particularly with the establishment of the microelectronics and materials handling research centers. But they still felt that the issue of academic interaction needs to be explored more aggressively.

Dr. George E. Dieter, Dean of Engineering, University of Maryland, commented that EES should attempt to "see what cooperation looks like from the academic side. . . until you know how they feel, you can't solve the problem." William R. Rambo, senior scientific advisor at SRI International, added that EES and the schools should capitalize on the resources they have to offer each other.

The group was pleased that EES had taken a "first cut" at creating a planning document, and recommended that we "keep at it" in a continuous refining process. They suggested that we focus on questions like these:

- What is our vision for the future? What and where do we want to be — and why?
- What should be our state and national role? Our relationship with industry?
- What are the strategic issues? What areas do we want to be #1 in and why? Looking at the areas we are working in now, which ones should be increased, decreased, or kept the same in emphasis?
- What should be our relationship with the academic side?
- How should career development and planning be handled for that essential element — people?
- What contingency plans should be made, in case of "surprises"?

They unanimously agreed that EES should change its name to one more descriptive of its mission. Dieter facetiously suggested: Georgia-Tech Research Institute for Technology and Science (GRITS)!

The advisors also urged EES to do political lobbying for measures, such as extension of the R&D tax credit, that affect its well-being.

Dr. David Morrison, President of IIT Research Institute, commented that the EES and individual lab plans are heavy on the quantitative aspects and the Station must deal with the qualitative aspects. "Are there market needs you want to serve?" he asked. "What will it take to do that?"

Dr. Charles M. Johnson, manager, Advanced Studies & Analysis Division, I.B.M., suggested three program areas to look at: areas of high national interest, possible spin-offs to make Georgia a higher technology state, and research areas industry won't handle.

William B. Leithauser, manager, Facilities Planning & Support Operation in General Electric's Major Appliance Business Group, stressed that if EES planned to grow by 15% a year, personnel training would be an essential ingredient for success.

Dr. Edward W. Ungar, director, Battelle Columbus Division, also attended the two-day meeting, but had to leave before the general debriefing.

Six of EES's external advisors watch Jim Hubbard of EMSL demonstrate the transmission electron microscope. Standing from left to right, they are: William Leithauser, George Dieter, William Rambo, Charles Johnson, Joseph Saloom and Edward Ungar. (Photo by Alan David)

A-3345

ENGINEERING EXPERIMENT STATION
Georgia Institute of Technology
A Unit of the University System of Georgia
Atlanta, Georgia 30332

~~IN CONFIDENCE~~

June 1, 1983

Captain Clark Myers
HQ Ballistic Missile Office
AFSC
Norton AFB, California 92409

Subject: M-X Deep Basing Heat Pipes for Thermal Dissipation
Monthly Progress/Status Meeting Report for
April, 1983

Dear Captain Myers:

This report summarizes activities on Research Project A-3345 (F33657-82-G-2083-R901) for the period 01 April 1983 through 30 April 1983. Planned activities for the period 01 May 1983 through 31 May 1983 are also presented.

1. Sink Heat Transfer Characteristics

A suitable numerical solution for 2-D (radial and circumferential) temperature profiles has been developed. This solution is based on superposition and takes into account interaction between the heat pipes. The attached graph is an example of the type analysis that can be performed using this model. The plot shows maximum heat transfer allowable per linear foot of 4" diameter heat pipe for a maximum surface temperature of 175°F at 10,000 hours. Once a complete set of heat pipe and header design equations have been developed, they can be used in conjunction with the rock temperature profile solutions to perform parametric study.

2. Heat Pipe and Waste Heat Removal Design

All steady state heat pipe equations including the gravity flow limitation, which also take into account fluid overfill, have been coded and the program debugged. Work continues directed towards development of transient heat pipe equations and coupling equations for the source and sink.

3. Heat Pipe Fabrication, Installation, and Maintenance

Continued work as described in the March monthly progress letter.

4. Planned Activities for June

Continue work as described above.

If there are any questions concerning this report or the status of the program, please call Walter Hendrix or Gene Colwell.

Respectfully submitted:

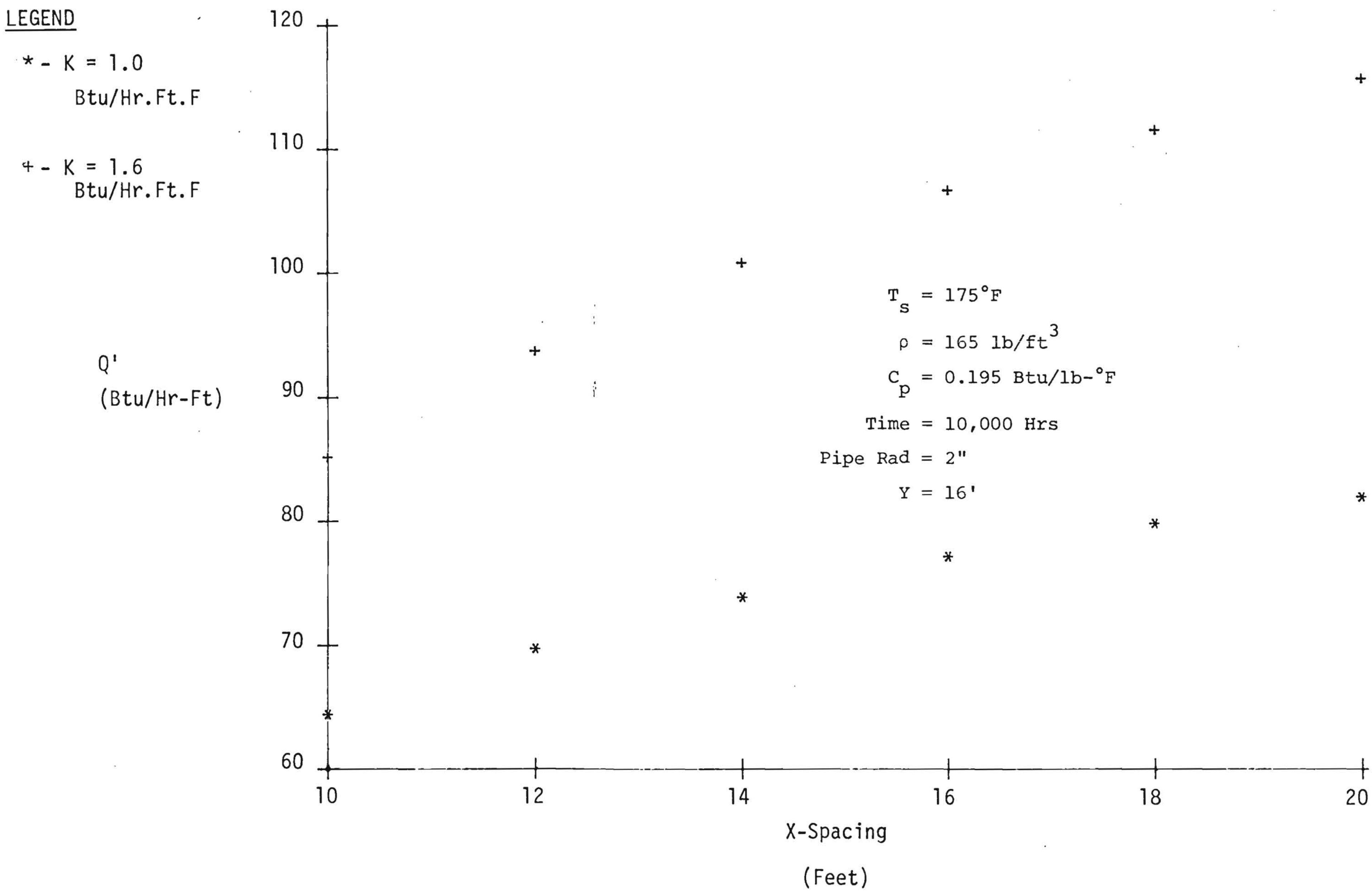
Walter A. Hendrix, P.E.
Co-Principal Invesitagator
Technology Applications Laboratory

Gene T. Colwell, P.E., PhD
Co-Principal Investigator
School of Mechanical Engineering

mro

Enclosure

MAXIMUM HEAT INPUT PER FOOT OF PIPE LENGTH



FINAL REPORT

~~IN CONFIDENCE~~

M-X DEEP BASING HEAT PIPES FOR THERMAL DISSIPATION

Phase 1 – Application Study

By

Walter A. Hendrix, Co-Principal Investigator
Gene T. Colwell, Co-Principal Investigator
V. Wesley Pidgeon
Michael L. Brown
Julio A. Santander

Prepared for

HQ Ballistic Missile Office
United States Air Force
Norton AFB, California

December 1983

GEORGIA INSTITUTE OF TECHNOLOGY

A Unit of the University System of Georgia
Engineering Experiment Station
Atlanta, Georgia 30332

GT EES

1983

M-X DEEP BASING HEAT PIPES FOR THERMAL DISSIPATION

PHASE 1 - APPLICATION STUDY

FINAL REPORT

Project Team

Walter A. Hendrix, Co-Principal Investigator
Gene T. Colwell, Co-Principal Investigator
V. Wesley Pidgeon
Michael L. Brown
Julio A. Santander

Prepared for

HQ Ballistic Missile Office
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Norton A F B, California

GEORGIA INSTITUTE OF TECHNOLOGY
Engineering Experiment Station
Technology Applications Laboratory

December, 1983

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SUMMARY

One of the plans being considered by the United States Air Force for M-X missile survivability and endurance is housing them in a deep underground tunnel facility called a Deep Base. This facility would likely be required to operate in the pre-attack mode for at least ten years and for up to one year, independent of external support, after post-attack "button-up". Included within the complex would be tunneling machines that could dig out of the cavern in preparation for missile launch as well as launch control and life support systems for the missile crew.

The support personnel and equipment required for such a basing mode would result in substantial amounts of waste heat which would have to be rejected in some manner. This situation implies the need for a heat sink capable of reliable and independent operation for a minimum of a year. It is proposed that heat pipes may be used to couple Deep Base waste heat sources to the surrounding rock in order to use it as a heat sink.

Because of the promise of heat pipes as a means for utilizing the surrounding rock environment as a sink for dissipation of waste heat from M-X Deep Bases, the Technology Applications Laboratory of the Georgia Tech Engineering Experiment Station, in cooperation with the School of Mechanical Engineering, is performing a feasibility study of this concept for the Ballistic Missile Office of the United States Air Force. This effort is divided into two phases with Phase 1 being an application study and Phase 2 being a technology study. The results of Phase 1 are detailed here and include analysis of the thermal characteristics of the rock environment and first-cut conceptual design for specific Deep Base applications.

Thermal analysis of the rock environment has centered around determination of representative rock properties and identification of analytical expressions suitable for calculating rock temperatures as a function of time and distance from the heat pipe. Investigation of the litertature has resulted in a range of rock physical properties which are representative of the potential Deep Base sites and from this range appropriate values may be selected for use in parametric studies. At the same time, a set of relatively simple algebraic expressions has been found with

which one can accurately predict rock temperature profiles at all time values of interest. The rock physical properties and temperature profile expressions are necessary information required to perform the modeling and parametric studies which will lead to optimum designs for heat pipe waste dissipation systems for Deep Base applications.

Preliminary conceptual design was performed for several selected Deep Base applications. Efforts were focused at designing modules which would handle some discrete portion of a particular waste heat load since it was felt that such an approach would maximize system flexibility and result in greater reliability and survivability.

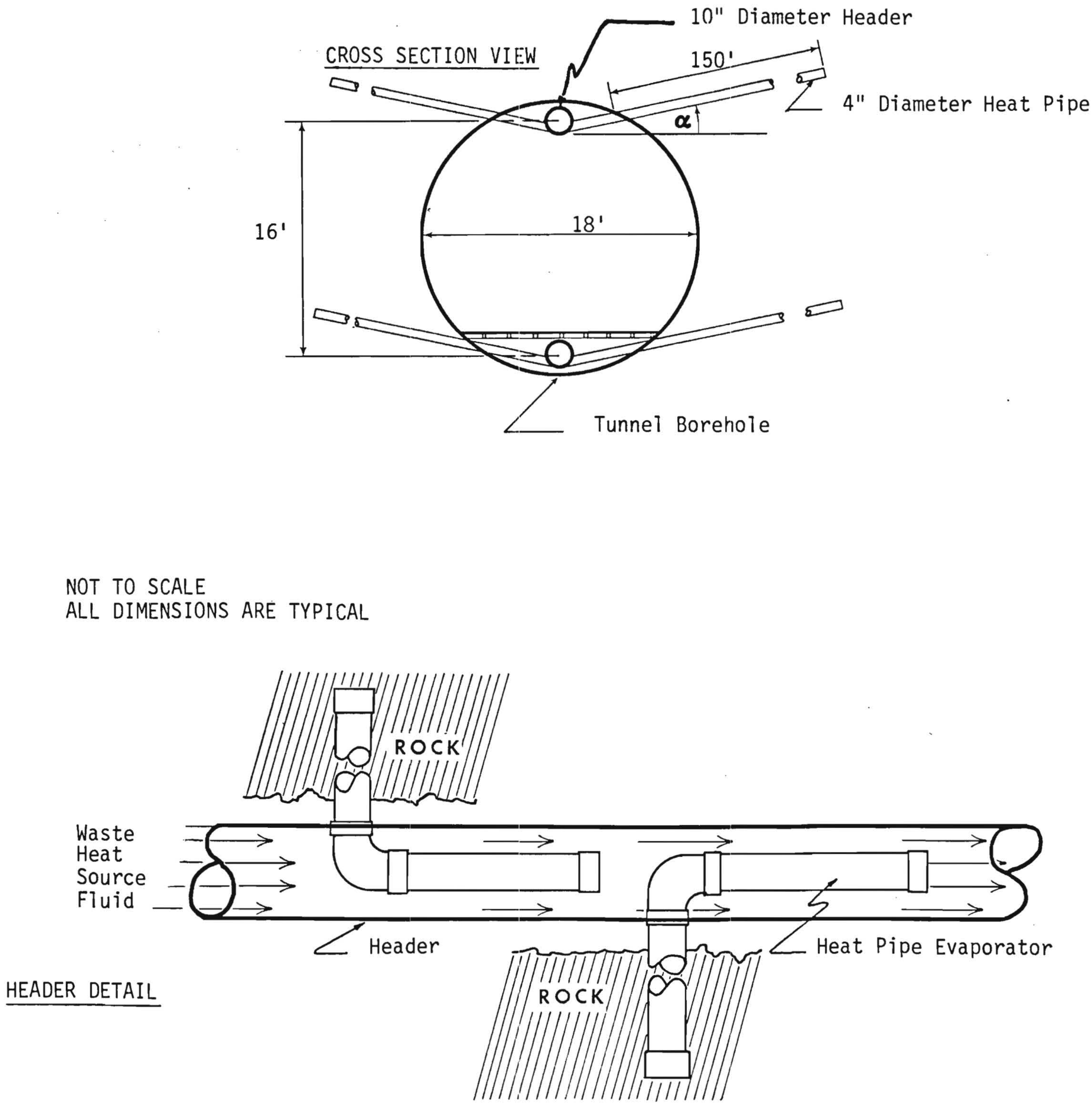
Removal of heat from air, water, and condensing steam or refrigerant was examined. Table 1 presents preliminary design of a one MWT module capable of handling any of these heat source fluids with appropriate modifications to the evaporator section of the heat pipes. In this preliminary design, stainless steel was used as the evaporator material, but most likely would not be considered in actual design since it is expensive and a relatively poor conductor. Instead, the material of choice would probably be plain carbon steel or copper. Figure 1 shows schematically a possible layout of heat pipes and headers in a tunnel. As presently conceived, the heat pipes would be axially staggered along the legnth of the header which would run along the top and bottom of the tunnel.

TABLE 1

PRELIMINARY MODULE CHARACTERISTICS

Module Size:	1 MWt
Heat Load per Heat Pipe:	5 KWt (17065 BTU/HR)
Heat Pipe O.D.:	4"
Heat Pipe Length (Condenser):	150'
Heat Pipe Length (Evaporator):	5'
Heat Pipe Material (Evaporator):	S. Steel
No. Heat Pipes per Module:	200
Total Header Length:	1000'
Tunnel Length Required:	500'

FIGURE 1. SCHEMATIC OF HEAT PIPE AND HEADER LAYOUT



The feasibility of using heat pipes to transfer heat generated by boring machines from tunnel air to the muck resulting from egress boring operations was evaluated. Heat transfer from tunnel air to the muck is essentially the same as the case considered earlier of removing heat from an air heat source and dumping it into the surrounding rock, so there are no inherent thermal problems with this concept. However, because of the required orientation of the heat pipe condenser section relative to the evaporator section, there may be practical problems with this application in boring tunnels which are vertical or nearly so.

Cooling of electronic equipment was also considered and appears, in general, to be a very good Deep Base application of heat pipes. Based on a search of the literature, several successful applications of heat pipes for this purpose may be cited (6, 7, 8). This concept should be further refined based on characteristics of expected electronic heat loads.

Preliminary investigation of materials which could be used to fabricate heat pipes (condenser and adiabatic section) was made. Because the poor thermal conductivity of the rock is the controlling factor in overall system performance, the thermal conductivity of the heat pipe material is less important relative to other issues such as survivability, ease of installation and maintenance, life, and cost. Based on these factors, a preliminary list of candidate materials has been formulated which ranges from very flexible elastomers through flexible but more rigid plastics and on to very rigid metals which still have a degree of flexibility. The choice of a flexible material may be very important to resisting shock and vibration, provide for easier installation, and possibly lead to lower cost if an inexpensive rubber or plastic can be used.

Even though the dimensions, configuration, materials and operating characteristics of the heat pipe waste heat removal systems considered here have not been optimized, they are feasible for all Deep Basing applications where the surrounding rock is used as the heat sink. Heat transfer from the tunnel air to the muck during egress boring operations can be accomplished using heat pipes in many instances and the use of heat pipes for electronic cooling appears, in general, to be a very good application. Therefore, this concept feasibility study should continue into Phase 2, the Technology Study, which will complete the analysis of the concept of using heat pipes to remove waste heat from Deep Bases and dissipate it in the surrounding rock environment. The complete study will culminate in conceptual designs for pipe heat dissipation systems for promising M-X Deep Base applications. This effort would also recommend laboratory and prototype system test programs which would address areas of technological uncertainty and lead to design information for manufacture, installation, operation, and maintenance of full-scale systems.

M-X DEEP BASING HEAT PIPES FOR THERMAL DISSIPATION
PHASE 1 -- APPLICATION STUDY

I. INTRODUCTION

One of the plans being considered by the United States Air Force for M-X missile survivability and endurance is housing them in a deep underground tunnel facility called a Deep Base. As presently envisioned, this concept would involve constructing a tunnel complex deep beneath the earth's surface in a geographic region of flat terrain with vertical walls such as a mesa. The facility would include tunneling machines that could dig out of the cavern in preparation for missile launch (egress) as well as launch control and life support systems for the missile crew.

The central underground complex would be a few thousand feet below the surface and could have multiple missile ports extending from the complex in a horizontal direction leading to the surface. Other tunnels might lead to openings at angles greater than horizontal. The tunnels could be dug with only a small portion left near the surface for the tunneling device to complete, or the machine might be capable of digging a new tunnel to the surface for missile launch.

The concept is aimed at maximizing hardness for M-X basing, to make it invulnerable to nuclear attack or to provide a favorable exchange in the event of an attack. Deep Basing in the early 1990's would provide enduring survivability and effectiveness against threats to deception and defense.

The support personnel and equipment required for such a basing mode would result in substantial amounts of waste heat which would have to be rejected in some manner. This situation implies the need for a heat sink capable of reliable operation for some specified period of time with a minimum of a year after "button up" being currently considered. Survivability constraints rule out communication with the earth's surface during the post-attack period which indicates that the heat sink must be the surrounding rock or else artificially created as part of the deep base facility (e.g., ice/water tunnels).

It is expected that there will be an increasing thermal gradient with depth in all of the potential basing areas so that the surrounding rock environment will be the same or higher than that desired for the tunnel facility.

In fact, this temperature difference could be substantial, on the order of 30-50^{o}F or more, depending on depth and geographic location. While one can insulate against heat gain from these high temperature surroundings, they present significant compatibility problems in attempting to utilize them as a heat sink for conventional heat rejection systems.

However, it appears plausible that the problems associated with the use of the rock surroundings as the heat sink for waste heat removal from M-X Deep Bases may be addressed effectively by the use of heat pipes to couple the heat source with the rock surroundings. Heat pipes would not require communication with the earth's surface to operate effectively and are passive devices which do not need a source of electrical or mechanical power. This passive characteristic is very important in that operation of the heat pipes would not draw power from the deep base energy system or add to the heat load that is being dissipated. Heat pipes do not require a large temperature difference across the condenser section in order to operate effectively. Therefore, the use of these devices would allow a relatively compact heat removal system since the condenser section would operate at temperatures only slightly higher than those in the surrounding rock. The heat pipes could be made of relatively inexpensive materials whose mechanical properties are tailored to resist shock waves associated with surface detonations or natural geological processes and could use an inexpensive and innocuous material, such as water or methanol, as the working fluid. The system could be installed such that the heat pipes were oriented at any angle between zero and ninety degrees with complete assurance that heat would be transferred only from the heat source to the surrounding rock, and never the reverse.

An array of heat pipes could be designed to give extremely good reliability. If desired, complete modules (including conventional heat removal equipment, such as commercial-size air conditioning units, integrated with an array of heat pipes acting as the condenser) could be utilized as a unit to further enhance reliability and facilitate maintenance. In this case, the loss from operation of one or two modules, for whatever reason, would have little effect on the system as a whole, and because of their simplicity could probably be easily repaired.

Because of the promise of heat pipes as a means for utilizing the surrounding rock environment as a sink for dissipation of waste heat from M-X

Deep Bases, the Technology Applications Laboratory of the Georgia Tech Engineering Experiment Station, in cooperation with the School of Mechanical Engineering, is performing a feasibility study of this concept for the Ballistic Missile Office of the United States Air Force. This study will analyze the thermal characteristics of the rock environment and delineate heat pipe design parameters such as size, geometry, orientation, capillary structure, working fluid(s), and operating temperature regimes. In addition, heat dissipation system characteristics such as number of pipes, configuration of heat pipe arrays, size and configuration of other equipment, and materials of construction will be evaluated.

System intallation techniques, maintenance requirements, and operating ranges and constraints will be determined. Areas of technological uncertainty will be identified and development steps to resolve these uncertainties recommended.

The complete study will culminate in a conceptual designs of a heat pipe heat dissipation systems for promising applications in M-X Deep Base facilities, and recommended laboratory and prototype system test programs which would lead to the design information required for building and operating full-scale installations. The effort is divided into two phases with Phase 1 being an application study and Phase 2 being a technology study. The results of Phase 1 are detailed in this report and include analysis of the thermal characteristics of the rock environment and first-cut conceptual design evaluations for specified Deep Base applications. These preliminary conceptual designs are to address development of rough configurations and dimensions, description of systems and their expected operating characteristics, order-of-magnitude cost estimates, and first-cut risk assessment.

II. DEEP BASE WASTE HEAT MANAGEMENT CHARACTERISTICS

The support systems for Deep Basing are power, environmental control, and life support (PECLSS) with waste heat management being one of the five major technical issues facing PECLSS. The recommended approach for waste heat management is to evaluate and adapt existing heat sink technology and facility design to Deep Basing. This concept feasibility study to evaluate the use of heat pipes for waste heat removal represents just such an approach.

No site has been selected for the Deep Base. Therefore, for the purposes of concept evaluation a typical test site is being used as a baseline site. The Deep Base has two fundamental modes of operation: the pre-attack mode which may extend over a period of ten years or more and the post-attack mode which begins when the facility is buttoned up in anticipation of a nuclear attack. After "button up", the facility is designed to operate for up to one year independent of any external support (endurance period). At any time during the endurance period, egress may be initiated with the dig-out operation requiring about 150 hours and launching of the missile an additional ten minutes.

For the pre-attack mode the power requirements are estimated to be 300 KW for each of three clusters based on 2 KW per person and 150 people per cluster living in a 70°F environment. For the post-attack mode, the power requirements are again estimated to be 300 KW per cluster for the endurance period of one year. An additional 1200 KW per cluster would be required for the 150 hours required for egress, and either 4500 KW per missile for horizontal concepts or 2600 KW per missile for vertical concepts for launch. The launch power is required for only ten minutes and may be provided by an on-board gas generator rather than from the base electric power supply. In the post-attack mode an 80°F environment will be maintained in each cluster.

During egress, heat will be generated by the tunnel boring operation. It is estimated that fifty percent of this heat will be transferred directly to the muck, or rock refuse, with the other half going into the tunnel air. It is proposed that equipment will be provided as part of the tunnel boring machine system to transfer the heat which goes into air to the muck.

Based on the above data and the anticipated requirements of PECLSS, this study will evaluate the following heat pipe application areas:

- Heat transfer to rock from hot water, air, or steam at temperatures up to 212°F at a power level of 1-10 megawatts thermal.
- Heat transfer to rock from a typical air conditioning refrigerant.
- Cooling of electronic equipment.
- Heat transfer from air to drilling muck for a temperature difference between the air and muck of 10-30°F and a power level of 300 kilowatts thermal.

III. THERMAL ANALYSIS OF SURROUNDING ROCK ENVIRONMENT

Thermal analysis of the surrounding rock environment has centered around two major issues: (1) determination of representative rock physical properties, and (2) identification of one or more analytical expressions which may be used to accurately calculate rock temperatures as a function of time and radial distance from the heat pipe. Both of these issues are extremely important to the performance of the parametric studies which will be required in order to analyze the various proposed heat pipe applications in which the surrounding rock is to be used as the heat sink.

Considerable investigation of the literature has focused at determination of representative rock physical properties to be used in performing analyses (1,2,3). Lack of specific information regarding the expected location of the Deep Base has resulted in the selection of a range of physical properties to be used. The ranges are given in Table 2 below, and are based on the anticipated geology of various potential Deep Base locations. Using guidance from continuing Deep Base studies, a set of representative rock properties will be selected from these ranges and used in all of the parametric studies.

The applicable governing differential equation (GDE) for analyzing the transfer of heat from the heat pipe into the rock involves the conduction of heat to a region bounded internally by an infinite circular cylinder. The simplest, but most conservative, case to analyze, that of one-dimensional radial heat flow to infinite surroundings at uniform initial temperatures, provides a suitable starting point for this study. This problem has been

TABLE 2.

RANGE OF REPRESENTATIVE ROCK PROPERTIES

Thermal Conductivity, BTU/hr-ft-F	0.5 - 2.5
Specific Heat at Constant Pressure, BTU/lb-F	0.15 - 0.25
Density, lb/ft^3	125 - 175

treated in a number of mathematical and engineering texts, notably that of Carslaw and Jaeger (4), giving rise to both general and closed-form solutions for temperature as a function of radial distance and time for the cases of constant cylinder surface temperature and constant cylinder heat flux. The constant heat flux case will be the most important in analyzing Deep Base heat pipe applications and the temperature profiles of most interest are those during the first few minutes after heat pipe start-up and after a year of operation.

The general solution of the GDE for constant heat flux is in the form of an infinite integral of an algebraic Bessel Function expression. This solution has been simplified by Carslaw and Jaeger to yield closed-form expressions for both small and large values of time. A different analytical approach which idealizes the problem as that of a continuous line source in an infinite medium also gives an expression which is relatively simple to evaluate. The GDE along with the general, closed-form, and line source solutions are presented in Appendix A.

The rock temperature profiles may also be solved by finite element analysis. Using this technique, the rock is divided into small elements and an energy balance is performed on each element. By taking suitably small elements and time increments, an accurate solution of rock temperature as a function of radial distance and time may be determined.

While numerical solutions could be used to accurately predict rock temperatures as a function of time and radial distance, this method requires a relatively large amount of computer time, particularly for calculating temperatures close to the pipe surface. Since a large number of parametric studies may be required in order to achieve an optimum design for each type of Deep Base waste heat dissipation system, use of a finite element analysis could result in excessive amounts of computer time. Therefore, the analytical solutions were evaluated to determine the range of accuracy and applicability so that they could possibly be used in modeling and parametric studies.

For simplicity in this phase of the feasibility study, thermal interaction between individual heat pipes will be neglected. This assumption may not render suitable accuracy for final design of heat pipe heat dissipation systems, but will result in a less complex treatment while providing for first-order analyses of systems performance. Interaction between heat pipes will be considered during Phase 2 of this feasibility study.

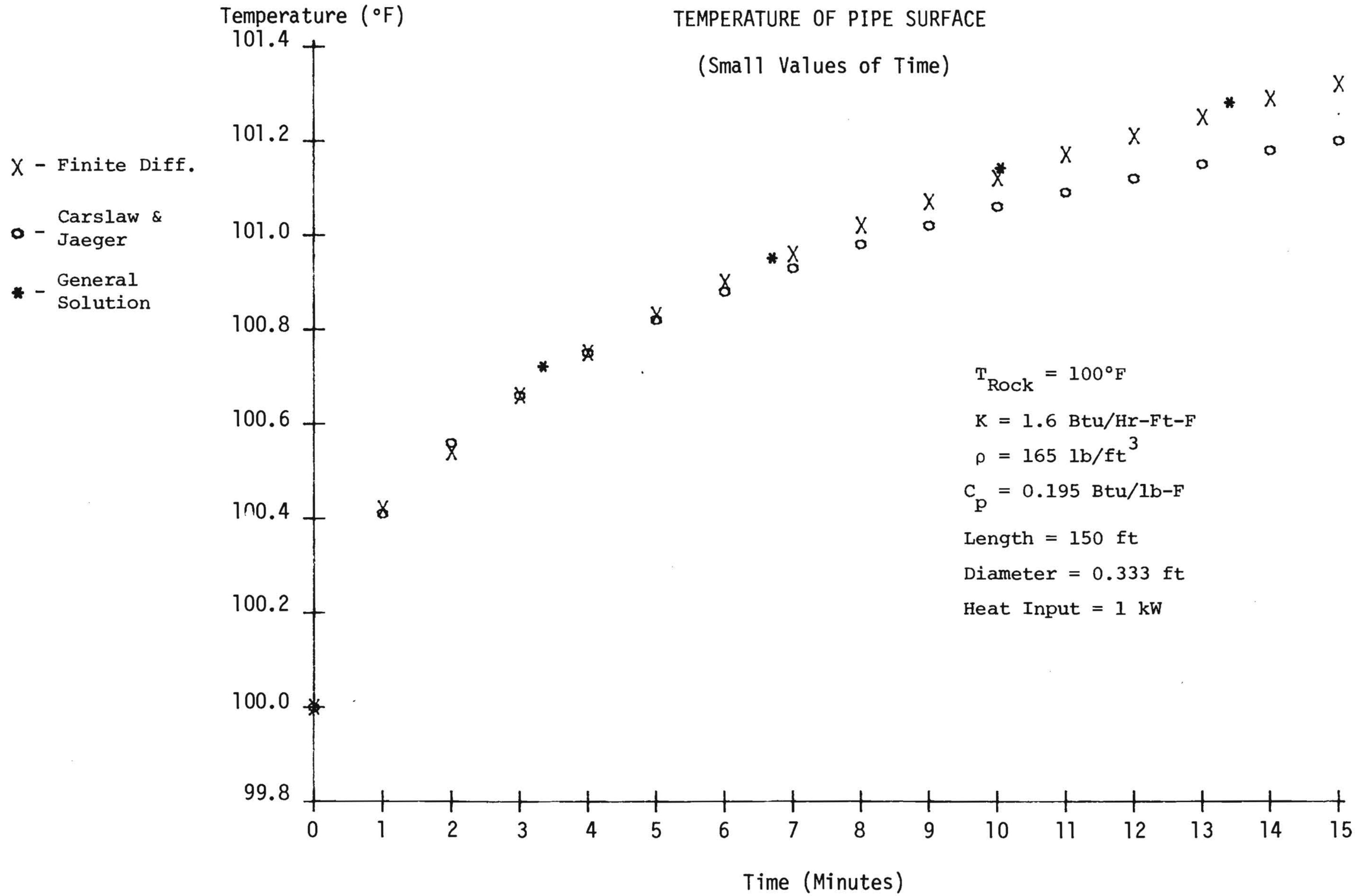
Figure 2 presents heat pipe surface temperature as a function of time for small values of time. The temperature values presented in this figure were obtained from finite difference solution, the Carslaw and Jaeger small time expression, and data from the literature calculated using the general solution (5). All three sources of data are seen to agree in the time interval 0-5 minutes. Therefore, the Carslaw and Jaeger small time expression can be used for initial modeling and parametric studies of heat pipe start-up. If a longer time period must be considered, for example up to 15 minutes, then finite element analysis must be used.

Figure 3 presents rock temperature as a function of radial distance from the heat-pipe surface at time equal to 10,000 hours. The temperature values presented in this figure were obtained from finite difference solution, the Carslaw and Jaeger large time expression, the line source expression, and the general solution literature data. In this case, all four sources of data agree very well up to approximately 25 feet from the heat pipe surface. Up to this distance the Carslaw and Jaeger large time expression can be used for initial modeling and parametric studies. At distances greater than 25 feet from the heat pipe surface the line source expression is preferred.

Using this set of analytical expressions for obtaining rock temperature profiles as a function of time and radial distance, one may obtain the data necessary to initiate the various parametric studies required in order to optimize the design of each type of Deep Base waste heat removal system. Figures 4, 5, and 6 illustrate the type of information which may be obtained.

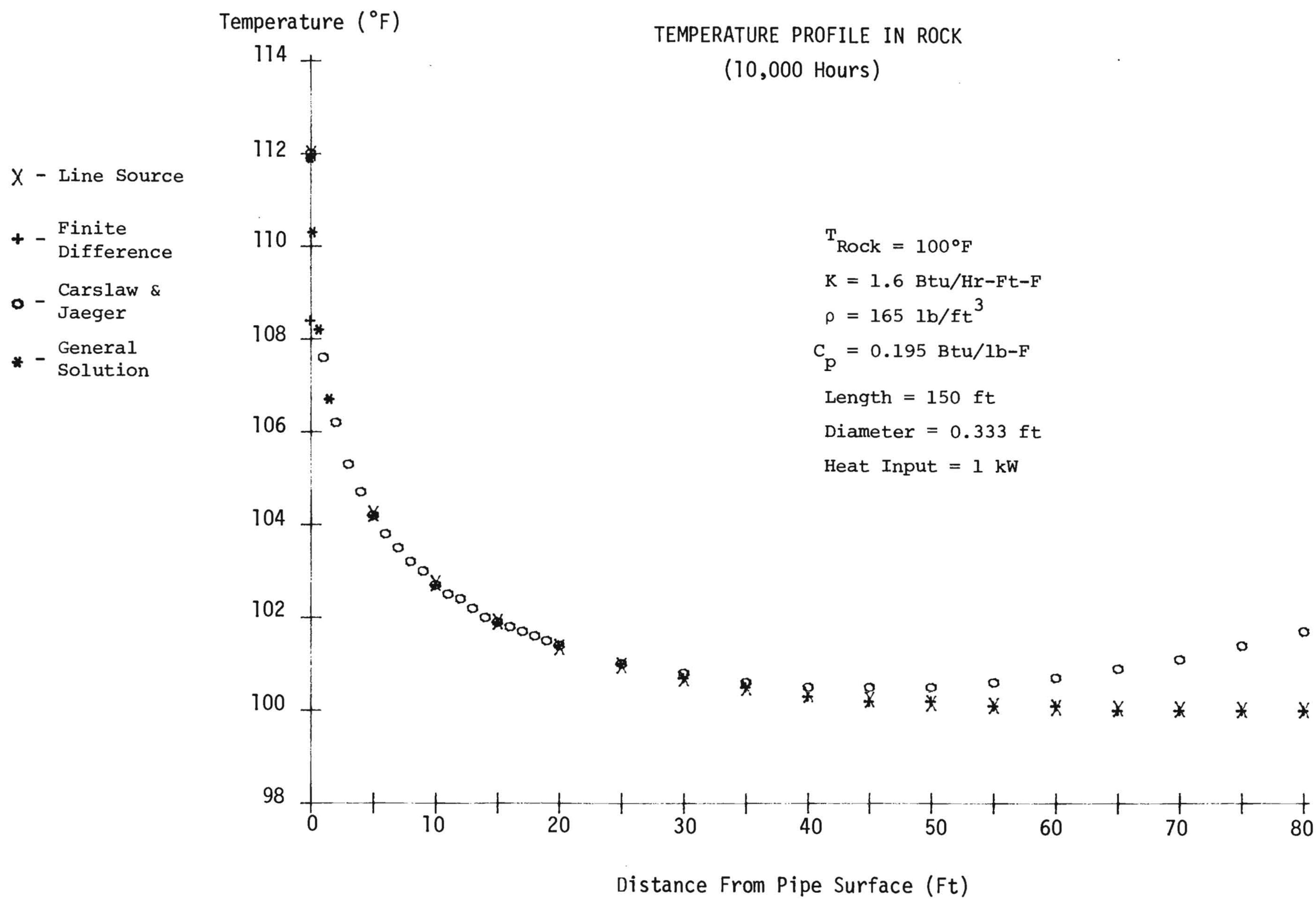
Figure 4 presents rock temperature profiles at time equal to 10,000 hours for a range of heat fluxes. This data allows one to study the effect of: 1) varying the heat load into the evaporator section of a heat pipe of constant length and diameter (in this case, 1, 3, and 5 KW/pipe at 150 ft pipe length), or 2) varying the length of the heat pipe while holding its diameter and evaporator heat load constant (in this case, 30, 50, and 150 ft pipe length at 1 KW/pipe).

Figure 5 presents rock temperature profiles at a time of 10,000 hours for various diameter heat pipes all having the same heat flux. This data illustrates the effect of: 1) varying the diameter of a constant length heat pipe which has a changing evaporator heat load, or 2) varying the diameter and length of a heat pipe with a constant evaporator heat load. It may not be



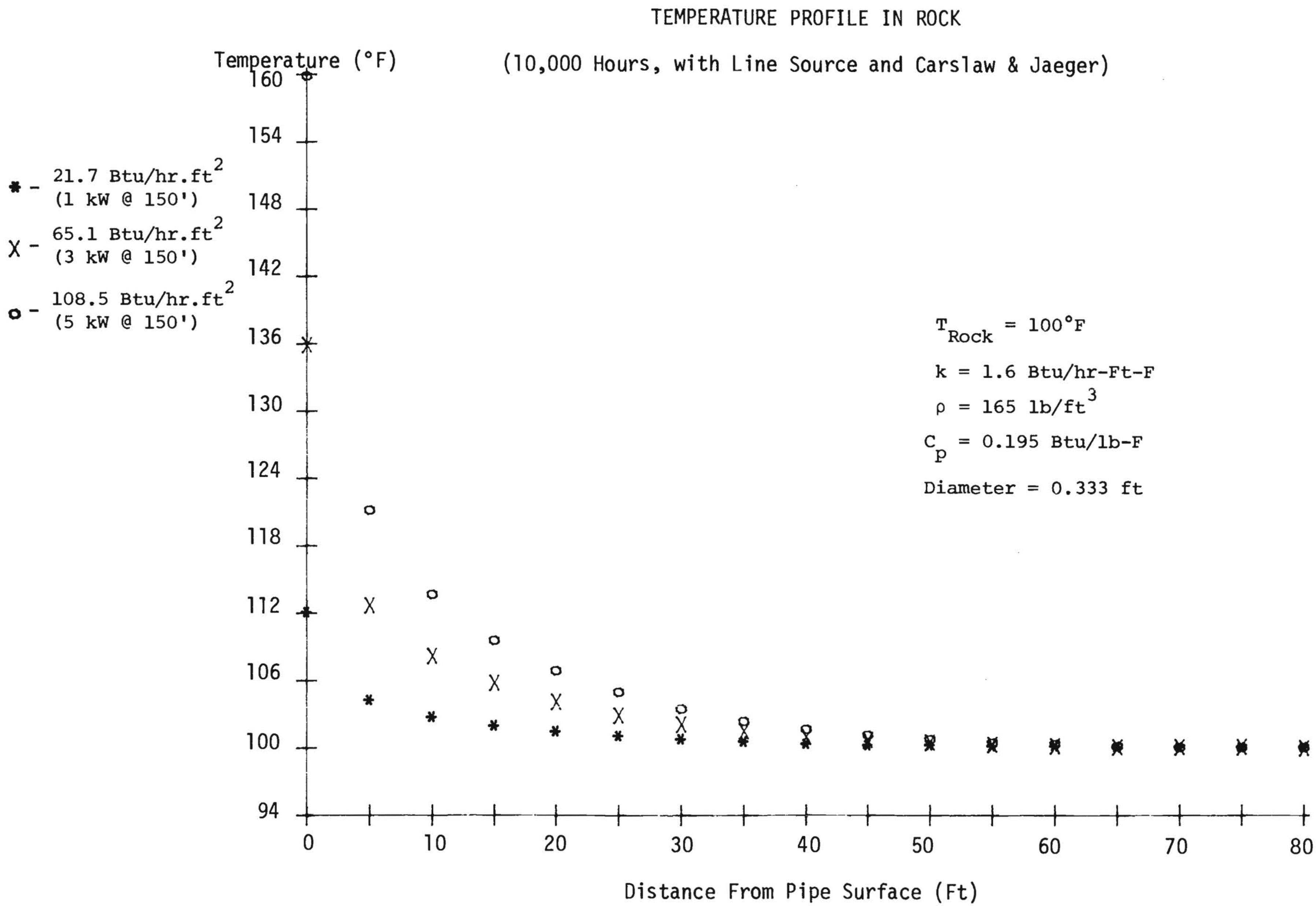
TEMPERATURE OF PIPE SURFACE
(Small Values of Time)
Temperature (°F)
101.4
101.2
101.0
100.8
100.6
100.4
100.2
100.0
99.8
X - Finite Diff.
o - Carslaw & Jaeger
* - General Solution
T_Rock = 100°F
K = 1.6 Btu/Hr-Ft-F
ρ = 165 lb/ft^3
C_p = 0.195 Btu/lb-F
Length = 150 ft
Diameter = 0.333 ft
Heat Input = 1 kW
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
Time (Minutes)

FIGURE 2



Temperature (°F)
TEMPERATURE PROFILE IN ROCK
(10,000 Hours)
114
112
110
108
106
104
102
100
98
X - Line Source
+ - Finite Difference
o - Carslaw & Jaeger
* - General Solution
T Rock = 100°F
K = 1.6 Btu/Hr-Ft-F
ρ = 165 lb/ft3
Cp = 0.195 Btu/lb-F
Length = 150 ft
Diameter = 0.333 ft
Heat Input = 1 kW
0
10
20
30
40
50
60
70
80
Distance From Pipe Surface (Ft)

FIGURE 3



TEMPERATURE PROFILE IN ROCK
(10,000 Hours, with Line Source and Carslaw & Jaeger)
Temperature (°F)
160
154
148
142
136
130
124
118
112
106
100
94
* - 21.7 Btu/hr.ft^2 (1 kW @ 150')
X - 65.1 Btu/hr.ft^2 (3 kW @ 150')
o - 108.5 Btu/hr.ft^2 (5 kW @ 150')
T_{Rock} = 100°F
k = 1.6 Btu/hr-Ft-F
ρ = 165 lb/ft^3
C_p = 0.195 Btu/lb-F
Diameter = 0.333 ft
0
10
20
30
40
50
60
70
80
Distance From Pipe Surface (Ft)

FIGURE 4

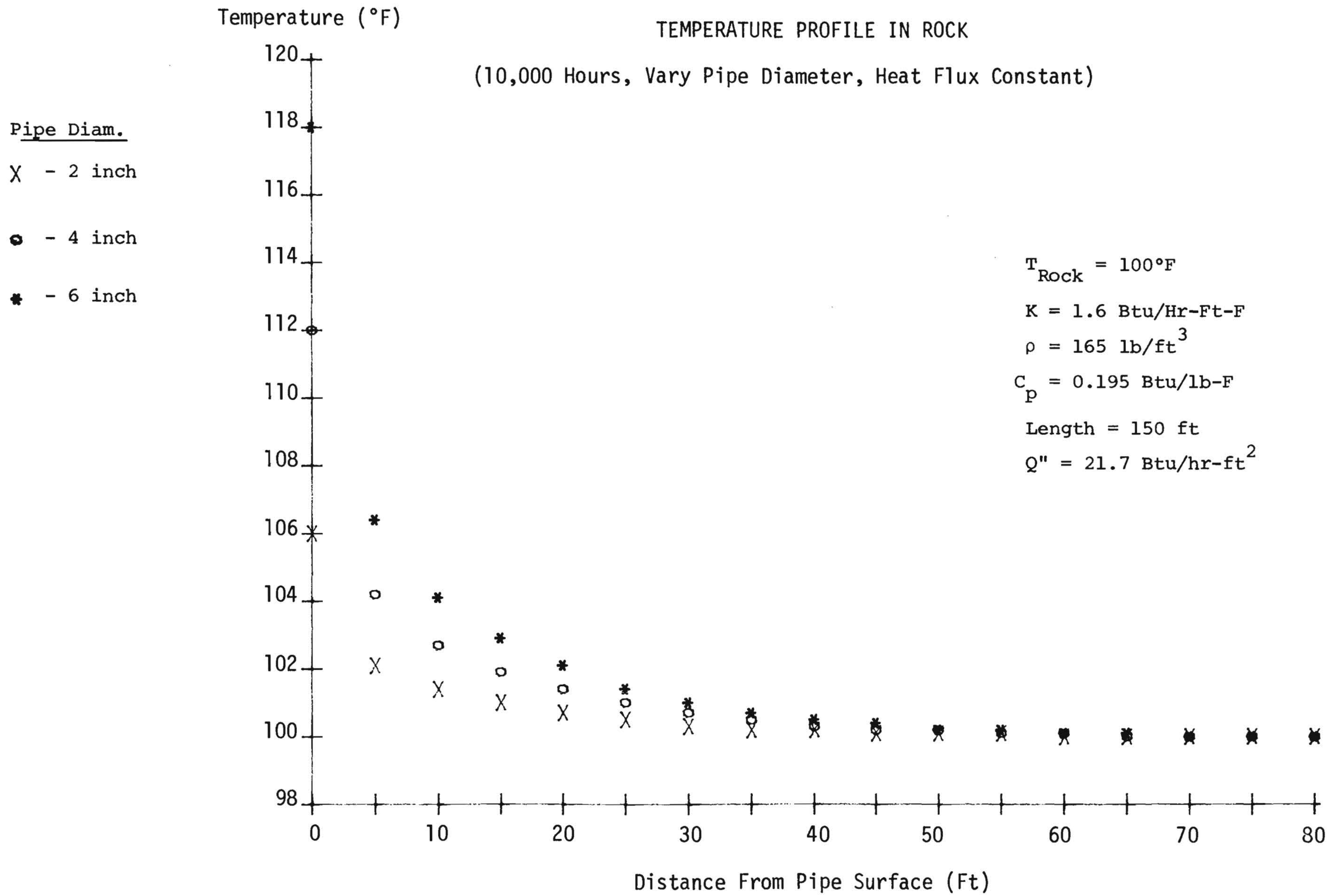

Temperature (°F)
TEMPERATURE PROFILE IN ROCK
(10,000 Hours, Vary Pipe Diameter, Heat Flux Constant)
Pipe Diam.
X - 2 inch
o - 4 inch
* - 6 inch
T Rock = 100°F
K = 1.6 Btu/Hr-Ft-F
ρ = 165 lb/ft3
Cp = 0.195 Btu/lb-F
Length = 150 ft
Q" = 21.7 Btu/hr-ft2
120
118
116
114
112
110
108
106
104
102
100
98
0
10
20
30
40
50
60
70
80
Distance From Pipe Surface (Ft)

FIGURE 5

intuitively obvious in Figure 5 why higher surface temperatures occur at constant heat flux for larger diameter pipes. This fact results, in the case of larger diameter, constant length pipes, because the rock volume available to absorb greater evaporator heat loads is constant. In the second case, that of larger diameter, shorter length pipes, higher surface temperatures occur because less rock volume is available to absorb the same evaporator heat load.

Figure 6 shows rock temperature profiles as a function of initial rock temperature for time equal to 10,000 hours and constant rock physical properties. These plots show the relative importance of Deep Base geographic location and depth which are factors related to initial temperatures.

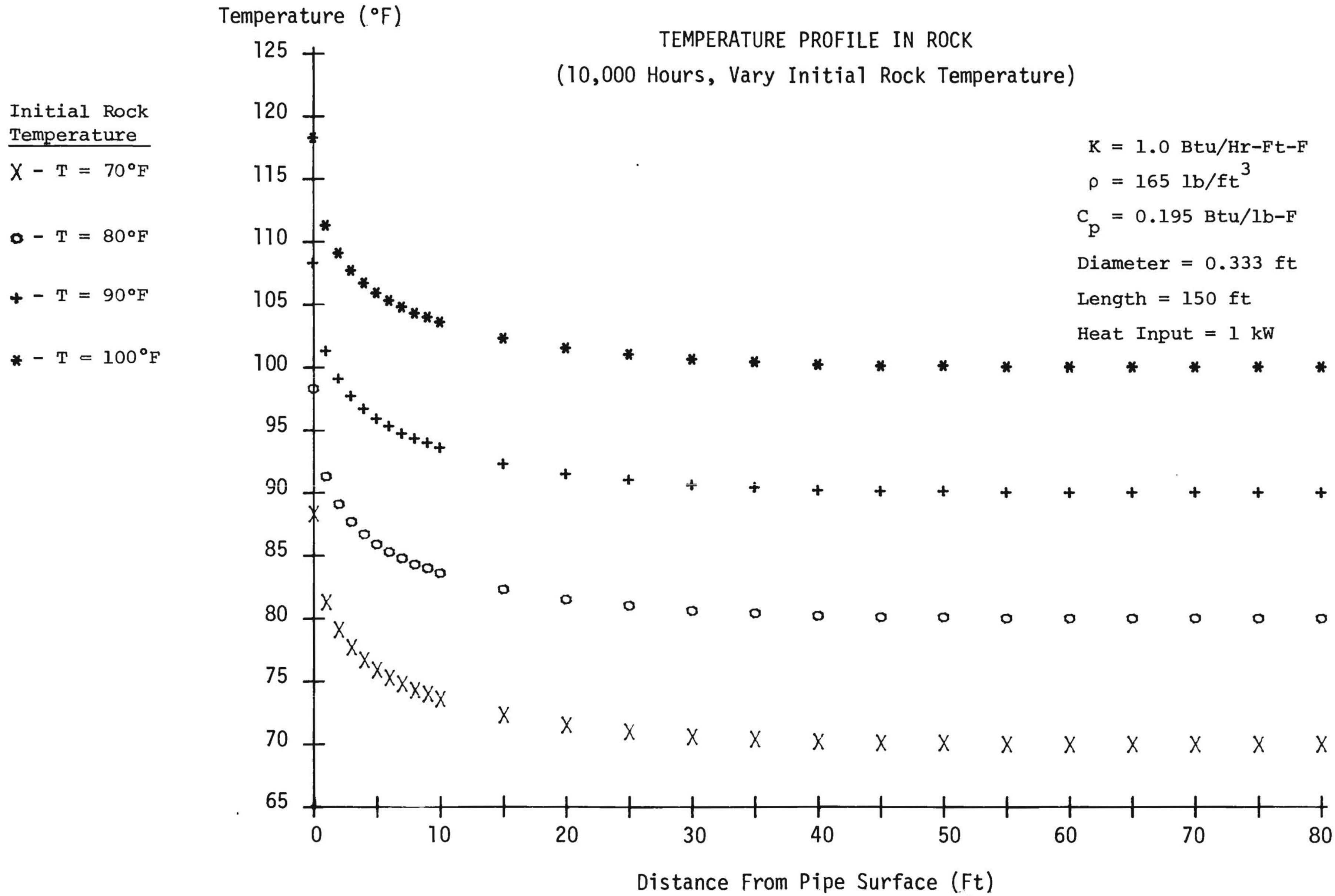

Temperature (°F)
TEMPERATURE PROFILE IN ROCK
(10,000 Hours, Vary Initial Rock Temperature)
Initial Rock Temperature
X - T = 70°F
o - T = 80°F
+ - T = 90°F
* - T = 100°F
K = 1.0 Btu/Hr-Ft-F
ρ = 165 lb/ft3
Cp = 0.195 Btu/lb-F
Diameter = 0.333 ft
Length = 150 ft
Heat Input = 1 kW
125
120
115
110
105
100
95
90
85
80
75
70
65
0
10
20
30
40
50
60
70
80
Distance From Pipe Surface (Ft)

FIGURE 6

IV. ANALYSIS OF SELECTED DEEP BASING APPLICATIONS

Conceptual designs of heat pipes and associated heat exchangers have been carried out for a variety of heat sources. These designs should be considered very preliminary since it is quite likely that geometries, materials, and operating conditions will be changed as the design optimization progresses and more is learned of the heat source equipment. Efforts were focused at designing modules which would handle some discrete portion of a particular waste heat load since it was felt that such an approach would maximize system flexibility and result in greater reliability and survivability.

Figure 7 shows schematically a possible layout of heat pipes and headers in a tunnel. As presently conceived, the heat pipes would be axially staggered along the length of the header which would run along the top and bottom of the tunnel. As indicated in Table 3, a one MWT module would require 200 heat pipes and a tunnel length of 500 feet. The fluid in the header may be air, water, condensing steam, or condensing refrigerant depending upon the source of heat. In this preliminary design stainless steel was used as the evaporator material, but, since it is expensive and also a relatively poor

TABLE 3.

PRELIMINARY MODULE CHARACTERISTICS

Module Size:	1 MWt
Heat Load per Heat Pipe:	5 KWt (17065 BTU/HR)
Heat Pipe O.D.:	4"
Heat Pipe Length (Condenser):	150'
Heat Pipe Length (Evaporator):	5'
Heat Pipe Material (Evaporator):	S. Steel
No. Heat Pipes per Module:	200
Total Header Length:	1000'
Tunnel Length Required:	500'

FIGURE 7. SCHEMATIC OF HEAT PIPE AND HEADER LAYOUT

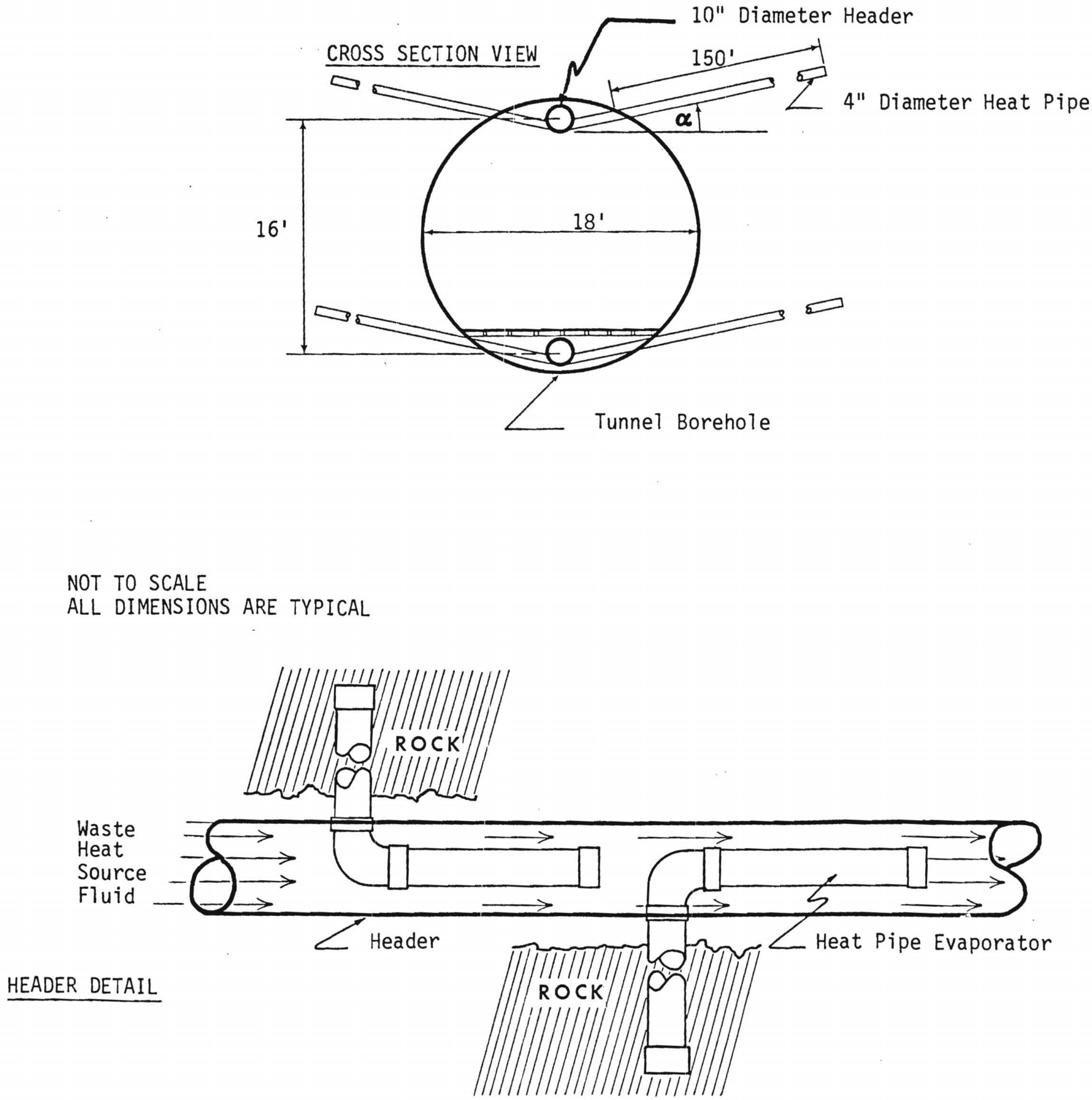


TABLE 4.

EVAPORATOR CHARACTERISTICS

$\dot{Q}_{in}$ = 5KWt (17065 $^{BTU}/_{HR}$)

Heat Source	(1) L_e (ft)	(2) $\dot{q}_e$ $\frac{BTU}{hr\ ft^2}$	(3) h $\frac{BTU}{hr\ ft^2\ {}^oF}$	(4) ΔT ^oF	V_{HF} ft/s	(6) $\dot{M}_{HF}$ Gal/min	(7) t_v ^oF	Heat Pipe Working Fluid
Water @ 212°F	5.00	3.26×10^3	176	18	(5) .47	114	181	Water
Condensing Steam @ 212°F	5.00	3.23×10^3	3230	1.0	48 (vapor in)	7.3 (liquid out)	199	Water
Condensing Freon-114 @ 200°F	9.00	(8) 1.84×10^3	123	15	7.6 (vapor in)	(8) 141 (liquid out)	173	Water
Water @ 212°F	5.00	3.26×10^3	176	18	(5) .47	114	171	Methanol
Condensing Steam @ 212°F	5.00	3.23×10^3	3230	1.0	48 (vapor in)	7.3 (liquid out)	189	Methanol
Condensing Freon-114 @ 200°F	9.00	(8) 1.84×10^3	123	15	7.6 (vapor in)	(8) 141	163	Methanol

(1) L_e = Evaporator Length

(2) $\dot{q}_e$ = Evaporator heat flux based on evaporator surface area.

(3) Film coefficient between header fluid & evaporator wall.

(4) Header fluid temp. minus evap. wall temp.

(5) Based on 150°F Exit Temperature (header fluid velocity)

(6) Header Fluid Flow rate

(7) Vapor temperature (heat pipe operating temperature) These values assume

(7) Con'd: that the Termal Resistance in the outside of the condenser is much larger than the rest of the thermal resistances of the systems.

(8) Actual heat load will be much smaller for an air conditioning system.

heat conductor, it would probably not be the material of choice in actual design. Instead, plain carbon steel or copper would probably be used.

Table 4 shows detailed results for preliminary design of the heat pipe evaporator section. The heat pipe working fluid (tube side fluid) is taken to be either water or methanol and the source fluid (shell side fluid) is taken to be liquid water at 212°F, condensing steam at 212°F, or condensing refrigerant-114 at 200°F. These calculations, which are based on axial shell side flow along the length of the evaporator tube, show that heat pipe evaporator lengths of 5 to 9 feet and temperature differences between outer evaporator wall and header fluid of 1° to 19°F are required. It would be possible to substantially reduce both the length and temperature differences by careful selection of header diameters and flow rates and by using simple heat transfer enhancement techniques.

When air is the heat source fluid, evaporator design requires more effort. As shown in Table 5, the film coefficient is quite low when parallel flow with an air velocity of 100 ft/sec is assumed. However, by using cross flow and fins a very respectable film coefficient of nearly 300 BTU/hr ft^{2}F can be achieved. There are many techniques for enhancing heat transfer for air flow on the outside of tubes and a method can be selected to yield required film coefficients which results in acceptable air side pressure drops.

TABLE 5.
AIR HEAT TRANSFER

Heat Source	Air Velocity Ft/s	h BTU/hr ft^{2}F Cross Flow No Fins	h BTU/hr ft^{2}F Cross Flow 16 fin/in	h BTU/hr ft^{2}F Parallel Flow Concentric HX
Air @ 200°F	100	15.24	292.4	21.8

Table 6 shows some overall characteristics for a heat pipe operating with the above mentioned heat source fluids and dumping the waste heat to a surrounding rock environment. Either water or methanol could be used as a working fluid depending upon heat source conditions and rock thermal characteristics. Both have been used extensively in heat pipes for many years and should present no unusual design, construction or operating problems. In this case, the material in the condenser section is assumed to be rubber or something similar to provide flexibility, which may be important to survivability and, possibly, lower cost. Overall temperature drops through the heat pipe of 40°F to 50°F are probably acceptable for the application being considered. However, they can be reduced considerably by careful selection of the flexible material used in the condenser section. If a metal condenser section material is chosen, then overall temperature drop through the heat pipe will in all cases be less than 10°F.

TABLE 6.
PRELIMINARY HEAT PIPE CHARACTERISTICS

Evaporator Length:	5'
Condenser Length:	175'
Adiabatic Length:	20'
Heat Pipe Material	
Evaporator	S. Steel
Condenser & Adiabatic Sect.	Rubber or Similar Material
Heat Pipe O.D.:	4"
Heat Pipe I.D.:	
Evaporator	3.5"
Condenser & Adiabatic Sect	3.2"
Capillary Structure:	100 Mesh (screen), 1 layer (S.Steel)
Working Fluid:	Water or Methanol
ΔT (Evap. Wall Temp - Cond. Wall Temp.):	
Water	40.19°F
Methanol	49.76°F
Heat Flux:	5 KWt (17065 $^{BTU}/_{HR}$)

The possibility of using heat pipes to transfer heat generated by boring machines from the tunnel air to the muck resulting from the egress boring operation has been considered. Assuming that the muck consists primarily of stone dust and chunks of rock and earth, then heat transfer from the heat pipe condenser surface will be similar to that into stone. Also, heat transfer between the evaporator surface and air will be governed by the same considerations as discussed previously for air flowing in a header. Therefore, no inherent problem exists for operation of heat pipes between air and drilling muck. However, the concept may be impractical because of the fact that heat pipe condenser surfaces must be at least slightly elevated relative to evaporator surfaces while egress tunnel boring may, at times, be vertical or nearly so making it difficult to achieve this required heat pipe orientation.

The feasibility of cooling electronic equipment using heat pipes was also investigated. There have been several successful applications of heat pipes in cooling electronic equipment. Nelson, et al developed circuit card heat pipes for electronic modules (6). The experimental results showed that the heat pipe had a better performance in lowering both the maximum component mounting surface temperature and the temperature gradient between components when compared with data for similar metallic thermal mounting plates. Additional work by Nelson, et al has been done in direct heat pipe cooling of semiconductor devices (7). Osakabe, et al successfully developed and mass produced several types of heat pipe heat sinks for cooling of semiconductor power devices of commercial audio power amplifiers (8). These heat pipe heat sinks were found to be about 30% more effective in radiator performance and about 50% lighter in weight than conventional heat sink made of extruded aluminum. These investigations indicate that cooling of electronic equipment in the Deep Base environment could be a very practical application.

V. HEAT PIPE MATERIALS

The controlling thermal factor in the use of any system, including heat pipes, for dumping heat to the surrounding rock is the heat transfer in the rock itself. Because this heat transfer is relatively poor, materials of construction may be selected for the heat pipes (condenser section) which have relatively poor thermal characteristics without adversely affecting overall thermal performance of the heat dissipation system. With thermal characteristics being less important, the major issues related to heat pipe materials selection are survivability, ease of installation and maintenance, life, and cost. Therefore, materials selection should focus on maximizing survivability, ease of installation and maintenance, and life while minimizing cost.

Further analysis will be required in order to completely delineate all of the important factors related to heat pipe survivability in Deep Base applications, but preliminary investigation indicates that length and flexibility should be considered. It appears that the shorter the heat pipes can be made without affecting system performace, the lower the risk of damage due to shock or vibration and the easier it will be to harden them against expected forces. On the other hand, degree of flexibility needed is not apparent at this time. Materials which are brittle should be avoided, but materials ranging from flexible and stiff to flexible and elastic may be considered.

Because of depth, size and specialized requirements of the Deep Base tunnel facility, large constraints will be placed on the installation and maintenance of the heat pipes. Further complexity will exist because of the requirements for high reliability and survivability. Selection of the heat pipe fabrication materials must take into account these factors in order to provide for ease of installation, system check-out, and in-place repair or replacement.

Expected life for the heat pipes must be greater than ten years since the Deep Base must be capable of operating in the pre-attack mode for at least this period of time. Primary factors related to heat pipe life are effects of chemical and thermal degradation and permeability of non-condensable gases. Chemical degradation may arise as a result of the working fluids or the rock environment. Though the source and sink temperatures being considered here are moderate, thermal degradation may occur for some candidate materials over a period of time. And finally, permeability to non-condensable gases must be minimized in order to minimize heat pipe length while assuring good thermal performance over their life.

Since the construction of Deep Bases is expected to be very expensive, providing a system for removing waste heat at the least possible cost is all the more important. Since a fairly large number of heat pipes will be required to handle all expected loads and provide some redundancy, the material chosen for fabricating the heat pipe will have some influence on total system cost.

Based on all the above mentioned aspects, a preliminary listing of representative candidate materials for heat pipe fabrication are given in Table 7. The materials under consideration range from very flexible elastomers which have low rigidity, through more rigid but still flexible plastics, and on to very rigid metals which still have a degree of flexibility. The thermal and mechanical properties presented in Table 5 represent those which are easily obtainable from handbooks and manufacturers of the various materials (9, 10, 11, 12, 13, 14, 15).

Flexible materials which will provide heat pipes which can be installed without undue complication and effort may be good candidates. Therefore, mechanical properties related to flexibility are listed for comparison.

All of the candidate materials exhibit good chemical resistance to both water and methanol up to and above expected operating temperatures and are relatively impermeable to gases. Most of these materials have experience with long term burial in soil and have shown good performance.

It is expected that the heat pipe capillary structure will be fabricated as an integral part of the heat pipe either through machining, forming, or etching the inside surface or providing a wick which plays a mechanical as well as thermal role. In addition, reinforcement of the basic material with a more rigid material or forming a composite of two or more materials to achieve a desired set of thermal, chemical and mechanical properties are also possibilities. Therefore, adhesion to metals or fabrics may be important properties.

TABLE 7. COMPARATIVE PROPERTIES FOR POTENTIAL MATERIALS FOR HEAT PIPES

	Hypalon[1] (chlorosulfonated Polyethylene)	Hytrel[1] (Polyester elastomer)	Nordel[1] (ethylene-propylene-diene polymer)	Vamac[1] (ethylene/ acrylic elastomer)	Polyethylene	Polypropylene	Teflon[1] (polytetrafluoroethylene)
Tensile Strength, psi	2500	6400	3000	2500	2000-4000	5000	2500-3500
Yield Strength, psi					-	-	-
Specific Gravity	1.12-1.28	1.20	0.83	1.08-1.12	0.94-0.96	0.89-0.91	21-2.3
Elongation, %	100-300	25 yield 600% break	100-300	100-300	25-400	500-700	250-350
Temperature Limit, ^{o}F (upper limit cont. service)	275	230	293	329	250	275-320	500
Thermal Conductivity	.08[(4)]	.08[(4)]	.08[(4)]	.08[(4)]	0.19[(4)]	0.08[(4)]	.14[(4)]
Permeability to Gases	Low- V.Low	Fair	Fair	V. Low	Low	Low	Low
Adhesion to Metals	Exc.	Good	Good-Exc.	Good	Poor	Poor	Good
Adhesion to Fabrics	Good	Outst.	Good	Excellent	Poor	Poor	Unknown
Tear Resistance	Fair	Outst.	Good	Good	Good	Good	Good
Abrasion Resistance	Exc.	Very Outst.	Exc.	Good	Good	Good	Good
Compression Set	Fair	Fair	Good	Good	-	-	-
Cost per Pound	$1.50	$2.75	$1.20	$2.20	$0.75	$0.75	$5.75
Additional Data	Thermoset	Thermo-plastic	Thermoset	Thermoset		Methanol resistance 140^{o}F-Satisfactory 212^{o}F-not suitable	

	Polybutylene	Ryton[2] (polyphenylene suffide)	Kynar[3] (polyvinylidene fluoride)	Copper 12200 DHP (deoxidized high Phosph. Alloy)	Aluminum 3003 - 0 (Soft tempered Al alloy)	Steel AISI 1010 HR
Tensile Strength, psi	3000-4500	9500	6500	32,000	16,000	47,000
Yield Strength, psi	-	-	-	9,000	6,000	26,000
Specific Gravity	.925	1.3	1.8	8.8	2.75	7.86
Elongation, %	200	-	25-500	40	30	28
Temperature Limit, ^{o}F (upper limit cont. service)	200	450	280^o (@ 220 psi)	400	400	-
Thermal Conductivity	.125 [(4)]	.165[(5)]	.065[(4)]	117[(4)]	224 [(4)]	25[(5)]
Permeability to Gases	Low	Low	Low	None	None	None
Adhesion to Metals	Unknown	-	Good	N/A	N/A	N/A
Adhesion to Fabrics	Good	-	-	N/A	N/A	N/A
Tear Resistance	Good	-	Good	Excellent	Excellent	Excellent
Abrasion Resistance	Very Good	-	Very Good	Excellent	Excellent	Excellent
Compression Set	-	-	-	N/A	N/A	N/A
Cost per Pound	$1.16	$4.05	$6.05	$1.40	$1.05	$0.50

(4) Units are BTU/hr ft$^{2\,o}$F Ft
(5) Units are BTU/hr ft^{o}F

[1]Trade name DuPont Company
[2]Trade name Phillips Petroleum
[3]Trade Name Pennwalt Corp.

And finally, the costs listed are those quoted by manufacturers, where available, on a per pound basis. Fabrication of tubing with a capillary structure and/or reinforcement a manufacture of composite tubing would, of course, result in higher costs. For example, rough estimates made using vendor data indicate that taking costs on a per foot basis would be approximately an order of magnitude greater than per pound costs.

VI. PLANNED WORK FOR PHASE 2 -- TECHNOLOGY STUDY

Even though the dimensions, configuration, materials, and operating characteristics of the heat pipe waste heat removal systems considered here have not been optimized, they are feasible for all of Deep Basing applications where the surrounding rock is used as the heat sink. Heat transfer from the air to the muck created during egress boring operations is feasible for some tunnel inclinations (i.e., those which are near or not too far from horizontal), but not for others (i.e., those which are vertical or nearly so). Application of heat pipes to electronic cooling is a very promising application and this concept should be further refined based on the characteristics of electronic heat loads.

The Phase 2 portion of this concept feasibility study will include continued heat sink analysis, heat pipe and waste heat dissipation system thermal and mechanical design for Deep Base applications, evaluation of system survivability, determination of installation, check-out, and maintenance requirements and procedures, and parametric study of heat pipe waste heat dissipation system performance. These studies shall address, but not be limited to, the following areas of interest:

A. Heat Sink Subsystem Design Considerations

(1) Layout of heat sink subsystem to include heat pipe/feed line interaction

(2) Modularity of design vs. response to attack

(3) Heat sink temperature profile/heat pipe interaction vs. time

(4) Parametric analysis of heat pipe performance vs. temperature gradient along feed line (inlet, midpoint, and outlet points)

B. Heat Pipe Design Considerations

(1) Materials available

(2) Estimates of material, fabrication, and assembly costs

(3) Preliminary heat pipe design drawings (sketches)/specifications

C. Heat Pipe Subsystem Installation and Checkout

(1) Methodology to insure "maximum" thermal contact at rock interface

(2) Methodology to perform heat pipe installation and checkout

D. Heat Pipe Subsystem Maintenance

E. Parametric study of heat pipe efficiency vs. variance in rock properties (k=1 and T_{rock} = 70^{o}-$100^{o}F$)

When completed, Phase 2 of this feasibility study, in conjunction with Phase 1 efforts which are detailed in this report, will comprise a complete analysis of the concept of using heat pipes to remove waste heat from Deep Bases and dissipate it in the surrounding rock environment. This completed study will culminate in a conceptual designs of a heat pipe heat dissipation systems for practical M-X Deep Base applications and recommended laboratory and prototype system test programs which would address areas of technological uncertainty and lead to design information for manufacture, installation, operation, and maintenace of full-scale systems.

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APPENDIX A

ROCK TEMPERATURE PROFILE EXPRESSIONS

Governing Differential Equation (1-D Radial Heat Flow)

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} - \frac{1}{\alpha}\frac{\partial T}{\partial t} = 0$$

General Solution

Constraints for Equation:

Q = Constant at $r = a$

$T_{pipe} = T_{rock}$ at time = 0

$$\theta = \frac{-2Q}{\pi k} \int_0^\infty (1 - e^{-\alpha u^2 t}) \frac{J_0(ur)\, Y_1(ua) - Y_0(ur)\, J_1(ua)}{u^2 [J_1^{\,2}(ua) + Y_1^{\,2}(ua)]} \, du$$

Solution for Small Time Valves

$$\theta = \frac{2Q}{k} \left(\frac{\alpha a t}{r}\right)^{\frac{1}{2}} \left[\text{ierfc} \frac{r-a}{2(\alpha t)^{\frac{1}{2}}} - \frac{(3r+a)(\alpha t)^{\frac{1}{2}}}{4\, ar} i^2\text{erfc} \frac{r-a}{2(\alpha t)^{\frac{1}{2}}} + \ldots \right]$$

Solution for Large Time Valves

$$\theta = \frac{Qa}{2k} \left[\ln \frac{4\alpha t}{Cr^2} + \frac{a^2}{2\alpha t} \ln \frac{4\alpha t}{Cr^2} + \frac{1}{4\alpha t}(a^2 + r^2 - 2a^2 \ln \frac{a}{r}) + \ldots \right]$$

Line Source Solution

$$\theta = \frac{Q'}{2\pi k}\ I\ (X)$$

Where: $I\ (X)$ = Exponenitial Integral = $\int_X^\infty \frac{e^{-\beta^2}}{\beta} d\beta$

$X = r/2\sqrt{\alpha t}$

NOMENCLATURE

a = radius of heat pipe, ft

α = thermal diffusivity = $k/\rho\ C_p$, ft^2/hr

C_p = specific heat at constant pressure, BTU/lb-F

k = thermal conductivity, BTU/hr-ft^2-F

Q = heat flux, BTU/hr-ft^2

r = distance from centerline of pipe, ft

t = time since start-up, hours

T = temperature at any point, °F

T_o = initial rock temperature, °F

θ = $T - T_o$, °F

γ = ln C = 0.57722 = Euler's Constant, Dimensionless

β, u = Variables of integration

J_o, J_1, Y_o, Y_1 = Bessel functions

ierfc = inverse error function

Q' = heat input per unit length of pipe, BTU/hr-ft

DATA FOR FIGURE 2

Time (Min)	Temperature (°F) FD	C & J	GS
0	100.00	100.00	
1	100.42	100.41	
2	100.54	100.56	
3	100.66	100.66	
3.352	-	-	100.72
4	100.75	100.75	
5	100.83	100.82	
6	100.90	100.88	
6.703	-	-	100.95
7	100.96	100.93	
8	101.02	100.98	
9	101.07	101.02	
10	101.12	101.06	
10.055	-	-	101.14
11	101.17	101.09	
12	101.21	101.12	
13	101.25	101.15	
13.407	-	-	101.28
14	101.29	101.18	
15	101.32	101.20	

DATA FOR FIGURE 3

Distance (Ft)	Temperature (°F) LI	FD	C & J	GS
0	112.00	108.4	112.0	111.9
0.166				110.3
0.666				108.2
1			107.6	
1.50				106.7
2			106.2	
3			105.3	
4			104.7	
5	104.23	104.2	104.2	
6			103.8	
7			103.5	
8			103.2	
9			103.0	
10	102.74	102.7	102.7	
11			102.5	
12			102.4	
13			102.2	
14			102.0	
15	101.91	101.9	101.9	
16			101.8	
17			101.7	
18			101.6	
19			101.5	
20	101.36	101.4	101.4	
25	100.97	101.0	101.1	
30	100.69	100.7	100.8	
35	100.49	100.5	100.6	
40	100.34	100.3	100.5	
45	100.24	100.2	100.5	
50	100.16	100.2	100.5	
55	100.11	100.1	100.6	
60	100.07	100.1	100.7	
65	100.05	100.0	100.9	
70	100.03	100.0	101.1	
75	100.02	100.0	101.4	
80	100	100.0	101.7	

Legend:

- FD - Finite Difference
- LI - Line Source Integral
- GS - General Solution
- C&J - Carslaw & Jaeger

DATA FOR FIGURE 4

Distance (Ft)	Temperature (°F)		
	1 KW	3 KW	5 KW
0	112.0	135.9	159.9
5	104.2	112.7	121.2
10	102.7	108.2	113.7
15	101.9	105.8	109.6
20	101.4	104.1	106.9
25	101.0	102.9	105.0
30	100.7	102.1	103.5
35	100.5	101.5	102.4
40	100.3	101.0	101.7
45	100.2	100.7	101.2
50	100.2	100.5	100.8
55	100.1	100.3	100.5
60	100.1	100.2	100.4
65	100.0	100.1	100.2
70	100.0	100.1	100.1
75	100.0	100.1	100.1
80	100.0	100.0	100.1

DATA FOR FIGURE 5

Distance (Ft)	Temperature (°F)		
	2" Pipe	4" Pipe	6" Pipe
0	106.0	112.0	118.0
5	102.1	104.2	106.4
10	101.4	102.7	104.1
15	101.0	101.9	102.9
20	100.7	101.4	102.1
25	100.5	101.0	101.4
30	100.3	100.7	101.0
35	100.2	100.5	100.7
40	100.2	100.3	100.5
45	100.1	100.2	100.4
50	100.1	100.2	100.2
55	100.1	100.1	100.2
60	100.0	100.1	100.1
65	100.0	100.0	100.1
70	100.0	100.0	100.0
75	100.0	100.0	100.0
80	100.0	100.0	100.0

FINAL REPORT
BMO/TR 84-07

DEEP BASING HEAT PIPES FOR THERMAL DISSIPATION

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FOREWORD

This Final Report was submitted by Georgia Institute of Technology, Technology Applications Laboratory, Engineering Experiment Station, Atlanta GA 30332, under Contract Number F33657-82-G-2083, with the Ballistic Missile Office, AFSC, Norton AFB, California. 2dLt Durrall Carroll BMO/SYBR, was the Project Officer in charge. This Technical Report has been reviewed and is approved for publication.

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

The Phase 1 work uncovered no insurmountable obstacles and, thus, work on the concept continued into Phase 2. The results of this effort, detailed in this report, are aimed primarily at specific technical issues which arose in the course of Phase 1 study. This work also provided a means for developing and validating the analysis and design tools needed to evaluate the thermal performance of a variety of Deep Base heat pipe applications. In addition,

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SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered) initial consideration of mechanical performance issues, such as survivability, installation, maintenance, and service life, are also included.

The Phase 2 work was essentially the next step in the process of validating the use of heat pipe technology for Deep Base waste heat removal. Such an effort required detailed development of all of the analytical expressions and numerical analysis techniques needed to design the waste heat removal systems and simulate their performance. Therefore, as an important part of this study, the classical analytical solutions for heat transfer in the rock were identified, the numerical techniques for analyzing cases such as thermal interaction between heat pipes were developed, and the design equations which will facilitate modeling of heat pipe performance were developed. These expressions and techniques were then coupled together in such a manner as to allow parametric study, preferably by digital computer of the performance of various heat pipe applications for Deep Base waste heat removal.

The results of this analysis shed some light on the value of the heat pipe/tunnel header system as a waste heat removal application for a Deep Base. However, they are primarily important as validation of the analytical procedures needed for thermal performance study of any Deep Base heat pipe waste heat removal application. These tools may be further refined, in cases where it is warranted, but in their present form they provide a firm basis for thermal analysis of the use of heat pipes to address specific waste heat removal problems that arise in the course of study of the Deep Base concept. Such study will address areas of technical uncertainty, leading to the information needed to overcome design and operational problems and make reasonable system cost estimates required to determine economic feasibility.

In conclusion, the concept feasibility study described in this report has shown that the use of heat pipes to remove waste heat from a Deep Base and dissipate it into the surrounding rock is a feasible and promising concept and, in fact, may be the best alternative to consider in some cases. In other instances, heat pipes might be effectively used to augment heat removal by alternate technologies, thereby improving the overall efficiency and reliability of Deep Base waste heat removal systems.

DEEP BASING HEAT PIPES FOR THERMAL DISSIPATION

PHASE 2 - TECHNOLOGY STUDY

FINAL REPORT
BMO/TR 84-07

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SUMMARY

The Air Force Ballistic Missile Office is actively studying the concept of the Deep Base as an alternative for storing and maintaining critical assets. This facility would consist of a tunnel complex buried thousands of feet below the surface of the earth equipped with all of the necessary systems for long term manned support and control for successful mission accomplishment. The Deep Base would likely be required to operate in a pre-attack mode, with normal access to surface facilities, for at least twelve and one half years, and for up to one year totally independent of external support after post-attack "button-up."

A major technological challenge for this concept is how to get rid of the large amounts of waste heat that would be generated by the necessary power plant, equipment, and people. Even the ambient rock environment at underground depths proposed for the Deep Base is not conducive to the concept, since it could easily have temperatures ranging from 70 to 100°F.

The Technology Applications Laboratory of Georgia Tech's Engineering Experiment Station, in cooperation with the School of Mechanical Engineering, is studying the feasibility of using heat pipes to dissipate waste heat from the Deep Base into the surrounding rock. Phase 1 of this concept feasibility study, completed in December, 1982, provided a first-cut analysis of applications of heat pipes for Deep Base waste heat removal. This effort was aimed at determining if there were significant technological or economic barriers which would preclude the use of heat pipes in Deep Bases.

The Phase 1 work uncovered no insurmountable obstacles and, thus, work on the concept continued into Phase 2. The results of this effort, detailed in this report, are aimed primarily at specific technical issues which arose in the course of Phase 1 study. This work also provided a means for developing and validating the analysis and design tools needed to evaluate the thermal performance of a variety of Deep Base heat pipe applications. In addition, initial consideration of mechanical performance issues, such as survivability, installation, maintenance, and service life, are also included.

A heat pipe is a passive, self-contained heat transfer device which has the capability of transferring large amounts of heat over small differences in temperature. The conventional form of this device is that of a closed container of arbitrary shape whose inside walls are lined with a capillary structure, or wick, and which is filled with a suitable volatile working fluid. The heat pipe operates by evaporating the working fluid at one point and condensing it at another, thereby utilizing the latent heat of vaporization of the working fluid as the means for storing heat that is to be transferred. This mechanism results in the high effective thermal conductivity and nearly isothermal operational characteristics of the heat pipe.

In order that the heat pipe evaporator not be depleted of working fluid, liquid must be returned from the condenser. This fluid movement is generally accomplished by capillary pumping action in the wick. An alternative mechanism for pumping the heat pipe working fluid back from the condenser to the evaporator is gravity flow. Heat pipes which operate on this principle are described as gravity-assisted and are the type envisioned for use in Deep Base waste heat removal applications.

These waste heat removal applications involve the installation of large, gravity-assisted heat pipes into the rock surrounding the Deep Base. Such systems would allow use of the rock environment as a heat sink in lieu of man-made, or artificially created, heat sinks; e.g., steam tunnels, ice/water tunnels. In each case, a heat source fluid would be circulated around the evaporator causing the working fluid in the heat pipe to evaporate, travel out to the condenser, where it would condense and give up its latent heat to the rock. The resulting liquid would then flow back to the evaporator by gravity where it would repeat the cycle.

Preliminary conceptual design of these Deep Base heat pipe systems was focused at designing modules which would handle some discrete portion of waste heat. For example, a module might be designed to handle one megawatt (MW) of power plant waste heat. In another case, a single module might provide cooling for a discrete portion of Deep Base equipment or section of the tunnel facility.

This approach was taken because it was felt that it would maximize system flexibility as well as provide for greater reliability and survivability.

Examples of Deep Base waste heat removal applications that have been identified thus far and appear to be attractive enough to merit continued study included:

- o Power Plant Waste Heat Removal
- o Equipment Cooling
- o Emergency Cooling

In addition to the stand-alone applications of heat pipes for Deep Base waste heat removal, heat pipes might also be coupled with other heat removal techniques to provide hybrid capability. Examples of hybrid systems, wherein heat pipes might fit, include such concepts as integration with ice/water tunnels, augmenting heat transfer inside steam or water tunnels, or back-up to other waste heat removal technologies to handle failures. The ability of heat pipes to provide transfer of heat at nearly isothermal conditions (i.e., over small temperature differences) and at low parasitic losses makes them excellent for enhancing heat removal by other techniques.

During Phase 1 of this study, a first-cut evaluation of the concept of using heat pipes for Deep Base waste heat removal was performed. Major areas of concern which were addressed included: (1) capability of the rock heat sink to absorb anticipated rates and quantities of waste heat, (2) preliminary heat pipe design to determine if practical geometries, working fluids, operating conditions and system configurations were possible, and (3) identification of areas of extreme vulnerability related to system reliability and survivability.

In each case considered, the heat pipes were spaced in such a manner as to be thermally non-communicating, which would be technically the most conservative, but not necessarily the most cost-effective, system. These heat pipe waste heat removal concepts, while preliminary and requiring additional technical and economic study, did not uncover any insurmountable barriers to implementation.

The Phase 2 work was essentially the next step in the process of validating the use of heat pipe technology for Deep Base waste heat removal. Such an effort required detailed development of all of the analytical expressions and numerical analysis techniques needed to design the waste heat removal systems and simulate their performance. Therefore, as an important part of this study, the classical analytical solutions for heat transfer in the rock were identified, the numerical techniques for analyzing cases such as thermal interaction between heat pipes were developed, and the design equations which will facilitate modeling of heat pipe performance were developed. These expressions and techniques were then coupled together in such a manner as to allow parametric study, preferably by digital computer of the performance of various heat pipe applications for Deep Base waste heat removal.

The efficacy of the above described analytical models was tested by using them for thermal performance analysis of the heat pipe/tunnel header waste heat removal concept. This proposed heat pipe waste heat removal system was conceived during the Phase 1 study and is described in detail in the introductory section of this report.

The results of this analysis shed some light on the value of the heat pipe/tunnel header system as a waste heat removal application for a deep base. However, they are primarily important as validation of the analytical procedures needed for thermal performance study of any Deep Base heat pipe waste heat removal application. These tools may be further refined, in cases where it is warranted, but in their present form they provide a firm basis for thermal analysis of the use of heat pipes to address specific waste heat removal problems that arise in the course of study of the Deep Base concept. Such study will address areas of technical uncertainty, leading to the information needed to overcome design and operational problems and make reasonable system cost estimates required to determine economic feasibility.

Though the Phase 2 study focused primarily on the thermal performance of heat pipe heat dissipation systems for Deep Bases, at least as important an issue is their mechanical performance. Major considerations related to mechanical performance are survivability, installation, maintenance, and service life.

Survivability of Deep Base heat pipe heat removal systems is related to mechanical failures due to causes other than those which might be considered normal for a system of this type; e.g., mechanical wear and tear. Failures of this nature would arise primarily as a result of the system being subjected to forces due to natural geological processes or weapons effects.

The factors expected to have the most influence on heat pipe survivability are geometry (heat pipe dimensions and system configuration), materials of fabrication, and redundancy. Geometry and materials of fabrication will be intimately related while redundancy refers to system overdesign to account for unexpected excursions and non-repairable loss from service of individual heat pipes or, perhaps, an entire heat pipe module.

A detailed study of the importance of geometry to survivability cannot be performed at this time because of its dependence on the thermal performance analysis, which is only in its initial stages, coupled with a lack of knowledge about the expected environment (i.e., types and magnitudes of mechanical forces). However, an example study was performed which analyzed the mechanical performance of a heat pipe when subjected to a particular geological event which could arise from natural or man-made forces. The analysis utilized GTSTRUDL, a computer-aided finite element structural engineering software system, to calculate the important stresses in the heat pipe wall resulting from the event.

The event analyzed in this example had the characteristics of minor block motion along a fault plane. This particular event was selected for an example because it could easily represent one of a number of events which could possibly occur as a result of stresses induced in the surrounding rock by the tunnel boring operation used to build the Deep Base facility.

This example study illustrated the analytical techniques needed to address the very complex issue of survivability. As the thermal performance analyses and other Deep Base systems studies progress, they will yield the specific information required to make a comprehensive survivability analysis of the heat pipe waste heat removal systems, using techniques such as GTSTRUDL.

System installation, performance monitoring, and maintenance are all important mechanical performance considerations. Because of the depth below ground level, limited size, and specialized requirements of the Deep Base tunnel facility, considerations related to these issues will be very important. Further complexity will exist because of the requirements for high reliability and survivability. These factors taken in combination result in several significant technical problems which must be studied and overcome in order to insure successful utilization of heat pipes in a Deep Base.

A survey of the state-of-the-art for long, horizontal drilling was performed. This study indicated that the technology for drilling the heat pipe installation holes is already in existence. In addition, horizontal drilling equipment is readily available which, if redesigned to satisfy the size and orientation constraints of the tunnel and system geometry, could be used to accomplish this task.

The most promising method for installation of heat pipes in a deep base is to insert them in sections. These sections could be joined to one another by threaded couplings, cementing, or welding, with any of these attachment methods capable of being performed automatically by machine.

Implicit in the installation concepts described above is the notion that the heat pipes will be fabricated in place. The entire installation procedure would entail a number of distinct steps. First, the installation hole would be drilled to the required depth and orientation at a specified tunnel location. Second, heat pipe condenser sections with internal capillary structure would be joined and inserted into the installation hole. Following insertion, the condenser would be cleaned thoroughly and plumbed to the evaporator which is expected to be fabricated outside of the Deep Base and then shipped in. The entire heat pipe would then be leak tested, filled with the prescribed amount of working fluid, and sealed.

Even though the thermal resistance of the air gap between the outer surface of the heat pipe condenser and the hole is expected to be small compared to the thermal resistance of the rock, it could still adversely affect overall performance of the heat removal system. If required, this problem could be overcome by displacing the air with a material which is a good heat conductor; i.e., a "thermal grout." This "thermal grout" would assure good heat transfer from the heat pipe to the rock.

Once all of the installation steps are completed, operation of an individual heat pipe could be tested by applying a known heat load to the evaporator and observing the ability of the heat pipe to efficiently remove the heat. In a similar manner the performance of entire heat pipe modules could be checked out and, in fact, the same technique could be used to monitor system operation, indicating deterioration in performance which might require maintenance.

Once a problem module is identified, individual heat pipes could be checked to determine which ones were operating poorly and causing performance loss. In many cases, problem heat pipes might be repaired simply by repeating some of the same steps which were used for initial installation. For example, a heat pipe could be disconnected at the evaporator and the fabrication steps of cleaning, evacuation, leak testing, and filling with working fluid repeated. In more extreme cases, the heat pipe condenser could be pulled from its installation hole and repaired or a new condenser inserted. For cases where the heat pipe is non-repairable, it might even be possible, depending on the location of the heat pipe in the tunnel facility, to drill an entirely new installation hole and refabricate a replacement heat pipe.

An important characteristic inherent in the heat pipe waste heat removal system being studied here is that the deterioration in performance or loss of a few of the heat pipes from service will not be detrimental to overall system performance. This characteristic is a result of the dispersed nature of this heat removal system and the fact that the design of the individual heat pipes can be made such that the operating heat pipes will pick up the increased heat load caused by the heat pipes which are out-of-service or operating with reduced efficiency.

A third mechanical performance area which is very important is system service life. Since the function of the Deep Base is to be a deterrent, the duration of this operational mode could be for its entire life which could easily be in excess of 20 years. Therefore, it is incumbent upon all of the Deep Base Systems, including those for waste heat removal, to have a service life of at least this duration.

Study of the factors related to heat pipe service life indicated that there are three non-attack related influences: thermal degradation, chemical degradation, and permeability to non-condensable gases or working fluid vapor. Fortunately, it was found that all three of these factors can be designed for fairly easily in Deep Base waste heat removal applications.

First, the expected operating temperatures for the heat pipes are moderate, since they will be on the order of 150-200°F. At these temperatures, thermal degradation of most candidate heat pipe materials would be mild, even over the extended timeframe of 20 years or more. Second, it is anticipated that the rock environment will be dry or contain, at most, only water, so that chemical degradation is expected to arise solely from attack by the working fluid or heat source fluid in the header. The header and working fluids which will be suitable for most of the Deep Base waste heat removal applications (water, methanol or, in special cases, freon or a freon mixture) are all relatively innocuous and not expected to cause significant chemical degradation, even over extended time periods, for most candidate heat pipe materials.

The final non-attack factor affecting heat pipe service life is permeability to gases and vapors. The presence of even minute amounts of non-condensable gases can severely reduce the film heat transfer coefficient in condensing processes. Permeability of condensable vapors is important because it could result in loss of working fluid.

Fortunately, however, the internal fluid dynamics of the heat pipe serves to minimize the effects of non-condensable gases and by careful design one may avoid significant deterioration in performance over the life of the system. Even in the event that a large build-up of non-condensables causes a reduction in heat pipe performance below that considered acceptable, remedial action may be taken by disconnecting the heat pipe at the evaporator and repeating the fabrication steps of cleaning, evacuating, leak testing, and charging with working fluid.

Permeation of the working fluid out of the heat pipe is relatively simple to avoid by judicious choice for materials of construction. For example, metals are essentially impermeable to gases and vapors whereas plastics and rubbers can be very permeable. Therefore, careful study of the physical properties of candidate materials is required before final material selection is made.

It should be noted that it is expected that the driving force for permeation, which is the pressure difference across the heat pipe wall, will be low throughout most of the life of the heat pipe. The maximum pressure difference across the heat pipe wall during both operational and idle periods, is expected to be no more than one atmosphere resulting in a relatively low tendency for non-condensable gases or working fluid vapors to permeate into the heat pipe.

Perhaps the most important result of all of the service life characteristics is that, barring catastrophic failure due to attack related phenomena, a heat pipe waste heat removal system will have graceful degradation in performance over the life of the Deep Base. The non-attack related factors affecting heat pipe service life are expected to result in minimal deterioration in performance and relatively few heat pipes lost totally from service. In addition, the operating heat pipes, by their very nature, will have the capability to compensate for loss from service or deteriorated performance of individual heat pipes so that effect on overall system performance will be negligible.

As indicated in the Phase 1 final report for this concept feasibility study, mechanical performance characteristics, as well as thermal performance characteristics, should be used as the basis for selecting the material for fabricating the heat pipe condenser. The controlling thermal factor for any Deep base heat removal system, including heat pipes, which dumps heat to the surrounding rock, is the heat transfer in the rock itself. Because this heat transfer is relatively poor, a material of construction may be selected for the heat pipe condenser section which has relatively poor thermal characteristics without adversely affecting overall thermal performance of the heat removal system. Therefore, potential heat pipe materials cover a broad range from very flexible elastomers through flexible but rigid plastics to very rigid, but not brittle, metals. In addition, it is possible that two or more of these candidate materials may be combined to form a composite which has the desired characteristics.

Detailed heat pipe waste heat removal system cost estimates are impossible to make, based on the results of Phase 1 and Phase 2 study alone, since both thermal and mechanical performance analyses are only in their initial stages. In fact, accurate assessment of system costs cannot be performed until the current study moves beyond the concept feasibility phase and specific Deep Base applications for heat pipes are chosen for evaluation. However, some preliminary cost implications based on knowledge gained thus far about potential systems were considered.

First, installation of heat pipe waste heat removal system are expected to require minimal lengths of dedicated tunnel. This characteristic is in contrast to some of the alternative Deep Base waste heat removal concepts being considered which require the boring of, potentially, miles of additional tunnel to house a heat sink (e.g., ice/water tunnels) or provide heat transfer surface for condensing the waste heat source fluid (e.g., steam tunnels).

Second, the drilling of small diameter holes only (relative to the anticipated tunnel diameter of 18 feet) will be required to install the heat pipes. The drilling of these holes would be based on existing horizontal drilling technology and would be accomplished using commercially available equipment adapted to meet the specialized constraints of the Deep Base. Costs for this type of drilling are expected to be greater than typical total costs for rock quarry drilling, but not a full order of magnitude greater.

Third, the materials being considered for use in the heat pipe waste heat removal systems are, in general, commercially available. Pipe made of these materials (common metals, plastics, or rubber) in the diameters being considered is an of-the-shelf item and can be joined in a variety of ways; e.g., threaded coupling, welding, cementing, etc. The fabrication of an integral capillary structure will, of course, add to the cost of this pipe, but this increase is not anticipated to be large.

Finally, study thus far has indicated that a relatively small number of heat pipes will be required to dissipate anticipated Deep Base waste heat loads. Preliminary calculations show that as few as 200 heat pipes might be required to remove one megawatt of waste heat. The number of heat pipes required will, of course, greatly influence the total cost of the waste heat removal system.

In conclusion, the concept feasibility study described in this report has shown that the use of heat pipes to remove waste heat from a Deep Base and dissipate it into the surrounding rock is a feasible and promising concept and, in fact, may be the best alternative to consider in some cases. In other instances, heat pipes might be effectively used to augment heat removal by alternate technologies, thereby improving the overall efficiency and reliability of Deep Base waste heat removal systems.

The next phase of study required for a complete program to validate the use of heat pipes in Deep Base waste heat removal applications would address system integration and optimization. This effort would investigate specifically how heat pipes might best be utilized for Deep Base waste heat removal. Actual applications of these heat pipe waste heat removal techniques would be analyzed as integral parts of the Deep Base waste heat removal system to determine their ability to provide for an optimum overall system. This analysis would utilize input on the nature and magnitude of expected waste heat loads, derived from other Deep Base systems studies, to perform modeling and parametric evaluations of thermal performance of potential heat pipe waste heat removal applications.

Following completion of the system integration and optimization study, steps could be taken to experimentally validate the use of heat pipe technology for Deep Base waste heat removal applications. The major development steps which would be required include: 1) testing of a single full-scale heat pipe in a laboratory, 2) testing of multiple, full scale heat pipes under conditions which simulate actual operation, 3) further study of waste heat removal system mechanical performance, and 4) final design and integration of practical Deep Base heat pipe waste heat removal applications.

This study has indicated that validation of the heat pipe waste heat removal concept for deep basing could be achieved by the end of 1986 if an aggressive program of study and testing, comprised of the technology development steps described above, are undertaken.

I. INTRODUCTION

Deep underground basing concepts have been studied for a variety of defense applications for a number of years. The United States Air Force considers the Deep Base a possible long-term solution to the problem of missile survivability and endurance -- for the 1990's and beyond. Additional applications of this technology might include deep underground command and control centers or bomb shelters to protect our nation's leaders.

The Deep Base would consist of a tunnel complex buried thousands of feet below the earth's surface suitable for the deployment of nuclear missiles. This facility would likely be required to operate in the pre-attack mode, with normal access to surface facilities, for at least twelve and one half years, and for up to one year totally independent of external support after post-attack "button-up." The complex would include a command and control center and support equipment for the missiles as well as tunneling machines which could possibly increase the size of the Deep Base or dig out in preparation for missile launch. Since it is planned that the facility will be manned by human crews it would also have to contain life support systems.

The Air Force Ballistic Missile Office is actively studying the concept of the Deep Base as an alternative for deploying nuclear missiles. A major technological challenge for this concept is how to get rid of the large amounts of waste heat that would be generated by the necessary power plant, equipment and people. Even the ambient rock environment at underground depths proposed for the Deep Base may have temperatures ranging from 70 to 100°F.

The Technology Applications Laboratory of Georgia Tech's Engineering Experiment Station, in cooperation with the School of Mechanical Engineering, is studying the feasibility of using heat pipes to dissipate waste heat from the Deep Base into the surrounding rock. Phase 1 of this concept feasibility study was completed in December 1982. This effort was an application study which provided a first-cut analysis of applications of heat pipes for Deep Base waste heat removal to determine if there were significant technological or economic barriers which would preclude their use. This work uncovered no insurmountable obstacles and, thus, work on the concept continued into Phase 2.

The Phase 2 effort is a technology study aimed primarily at addressing specific technical issues which arose in the course of the Phase 1 study. This work also provided a means for developing and validating the analysis and design tools needed to evaluate the thermal performance of a variety of Deep Base heat pipe applications. In addition, initial considerations of mechanical performance issues, such as survivability, installation and maintenance, and service life, are also included.

Of necessity, since data related to specific Deep Base waste heat removal applications have not yet been available, both thermal and mechanical performance studies during Phase 2 have been general in nature. However, the results of these studies have been sufficient to illustrate that the use of heat pipes for Deep Base waste heat removal is a very promising concept.

A. The Heat Pipe Waste Heat Removal Concept

A heat pipe is a passive, self-contained heat transfer device which has the capability of transferring large amounts of heat over small differences in temperature. The conventional form of this device is that of a closed container of an arbitrary shape whose inside walls are lined with a capillary structure, or wick, and which is filled with a suitable volatile working fluid. Due to a number of factors, such as ease of fabrication, availability of materials, and strength considerations, the most common form of heat pipe seen in actual practice is that of a right circular cylinder as shown in Figure 1. Many other geometries, however, are used in practical applications; e.g., flat plates, finned tubes, annular tubes, curved or flexible tubes.

The heat pipe operates by evaporating the working fluid at one point (the evaporator) and condensing it at another (the condenser), thereby utilizing the latent heat of vaporization of the working fluid as the means for storing heat that is to be transferred. This mechanism results in the high effective thermal conductivity and nearly isothermal operational characteristics of the heat pipe.

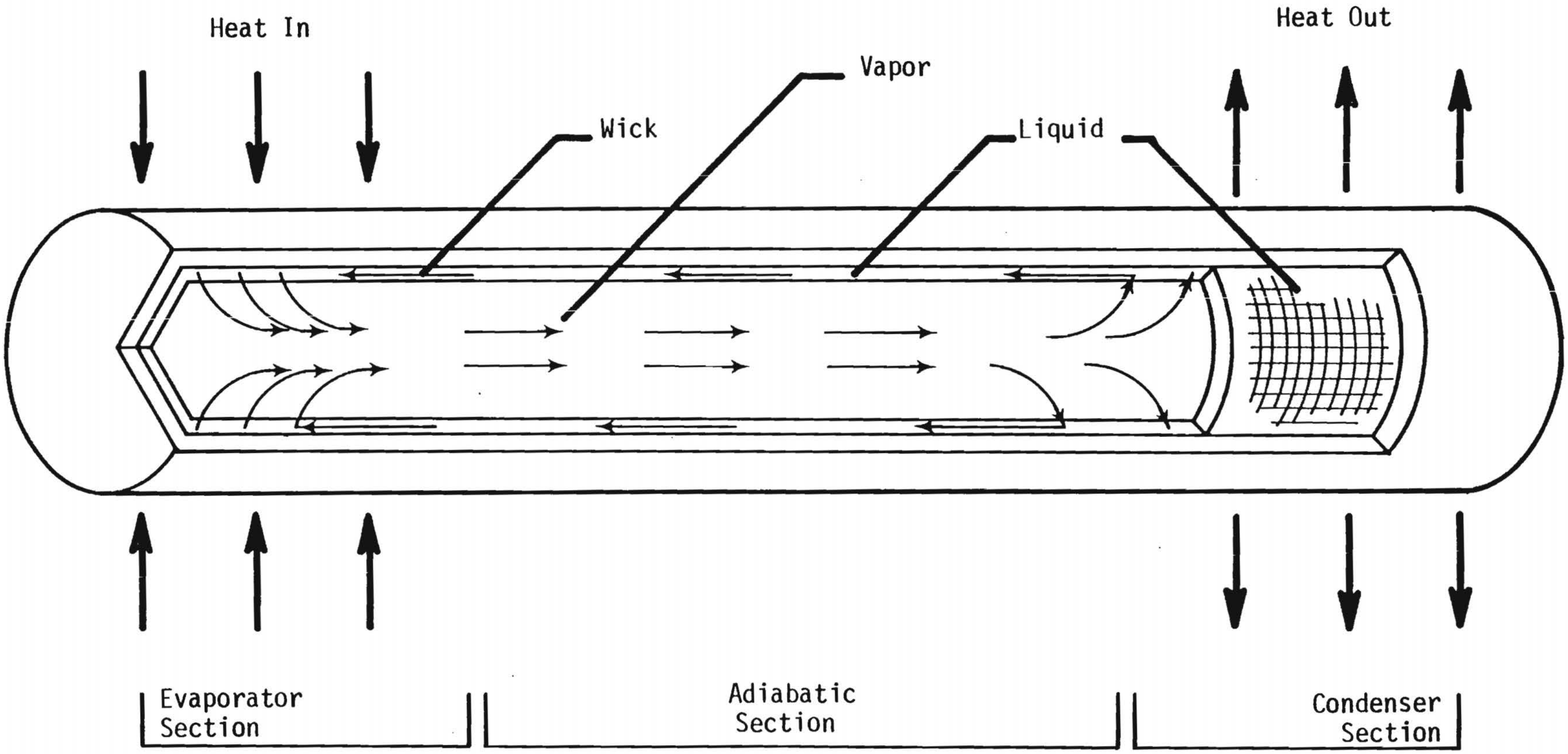


FIGURE 1

CUTAWAY OF A CYLINDRICAL HEAT PIPE SHOWING HEAT AND WORKING FLUID FLOWS.

In the conventional heat pipe, vapor flow is facilitated by the pressure difference created by the evaporating and condensing processes. In order that the evaporator not be depleted of working fluid, liquid must be returned from the condenser to the evaporator. This fluid movement is generally accomplished by capillary pumping action in the wick. These two fluid flow mechanisms allow the heat pipe to be operated in practically any orientation, even vertical with the evaporator above the condenser, without an auxiliary power source for pumping. However, the limit in capillary pressure difference that can be generated for a given capillary structure, being related to the difference in radii of curvature of the liquid-vapor interfaces in the evaporator and condenser, will affect heat pipe operational characteristics such as size, orientation, and heat transfer capacity.

An alternative mechanism for pumping the heat pipe working fluid back from the condenser to the evaporator is gravity flow. Heat pipes which operate on this principle are described as gravity-assisted and resemble, in many ways, a thermosyphon. In the thermosyphon a liquid boils in the lower regions of a container and condenses in the upper regions, giving up its latent heat, with the condensate returning to the boiler by gravity.

The capillary structure in the gravity-assisted heat pipe functions mainly to improve heat transfer by providing for better distribution of the working fluid. The pumping capability of gravity flow is much greater, in general, than that which can be obtained by capillary forces so that gravity-assisted heat pipes can typically be made much longer than conventional heat pipes. Gravity-assisted heat pipes also act as thermal diodes transferring heat in only one direction. This type of heat pipe does have the drawback that its installation must always be such that the condenser is elevated with respect to the evaporator.

Because of specific needs, a variety of heat pipes have been developed which utilize working fluids ranging from cryogenic liquids to liquid metals. These working fluids allow the heat pipe to become functional over a wide range of operating temperatures, from as little as 5 K to as much as 3000 K. The operating temperature is defined as the average temperature of the vapor phase when the heat pipe is operating normally. The internal physical process that occurs during operation of the heat pipe requires that its operating temperature lie between the freezing and critical temperature of the working fluid.

Heat pipe waste heat removal systems envisioned for Deep Bases involve the installation of long, gravity-assisted heat pipes into the rock surrounding the Deep Base. Such systems would allow use of the rock environment as a heat sink in lieu of man-made, or artificially created, heat sinks; e.g., steam tunnels, ice/water tunnels. In each case, a heat source fluid would be circulated around the evaporator causing the working fluid in the heat pipe to evaporate, travel out to the condenser, where it would condense and give up its latent heat to the rock. The resulting liquid would then flow back to the evaporator by gravity where it would repeat the cycle.

Preliminary conceptual designs of these Deep Base heat pipe systems were focused at designing modules which would handle some discrete portion of waste heat. For example, a module might be designed to handle one megawatt of power plant waste heat. In another case, a single module might provide cooling for a discrete portion of Deep Base equipment or section of the tunnel facility. This approach was taken because it was felt that it would maximize system flexibility as well as provide for greater reliability and survivability.

Examples of Deep Base waste heat removal applications that were identified in Phase 1 and appear to be attractive include:

> <u>Power plant waste heat removal</u>. The system design for removal of power plant waste heat could be very flexible to accommodate discrete portions of the load (1-10 MWT using 1 MWT modules has been considered thus far). The system would be inexpensive when compared to other methods since the need for miles of dedicated tunnel would be obviated and the required number of heat pipes to handle maximum load is relatively small.
>
> <u>Emergency cooling</u>. An example of this application would be nuclear reactor core cooling. A heat pipe system would allow direct removal from the core to the rock, thereby minimizing

the requirement for heat exchangers, pumps, and piping. The system could be designed to operate automatically in case of emergency shutdown or manually for planned outages and would utilize the rock heat sink at "virgin" or "near virgin" conditions when heat removal capability would be greatest.

Cooling of equipment. Examples of this application are cooling of electrical and electronic equipment, environmental control and life support processes, and similar sources of unwanted waste heat. Use of a heat pipe system in these cases would allow removal of the waste heat directly from the source to the rock heat sink before it has a chance to degrade to ambient temperature resulting in additional air conditioning load. Heat pipes could accomplish this function in a passive manner which eliminates the need for extensive heat exchange and pumping equipment as well as complicated and lengthy piping. Parasitic losses would also be minimized and, in cases where some pumping of the source fluid around the heat pipe evaporator is required, the prime mover could be hermetically sealed in the heat source fluid so that the heat pipe could also remove heat from frictional losses in the prime mover.

In addition to the specific applications of heat pipes for Deep Base waste heat removal described above, heat pipes might also be coupled with other heat removal techniques to provide hybrid capability. Examples of hybrid systems, wherein heat pipes might fit, include such concepts as integration with ice/water tunnels, augmenting heat transfer inside steam or water tunnels, or back-up to other waste heat removal technologies to handle failures. The ability of heat pipes to provide transfer of heat at nearly isothermal conditions (i.e., over small temperature differences) and at low parasitic losses makes them excellent for enhancing heat removal by other techniques.

B. Feasibility Study Approach

In order to firmly establish the feasibility of the use of heat pipes for Deep Base waste heat removal, there are several technical areas which must be investigated. These areas are primarily related to the heat transfer characteristics of the rock heat sink, thermal design of the heat pipes, and thermal and mechanical performance of the heat pipe waste heat removal systems.

During Phase 1 of this study, a first-cut evaluation of the concept of using heat pipes for Deep Base waste heat removal was performed. Major areas of concern which were addressed included: (1) capability of the rock heat sink to absorb anticipated rates and quantities of waste heat, (2) preliminary heat pipe design to determine if practical geometries, working fluids, operating conditions and system configurations were possible, and (3) identification of areas of extreme vulnerability related to system reliability and survivability. This effort utilized known analytical expressions for heat transfer in the rock and known heat pipe design equations to consider the dissipation of the following variety of heat loads:

- Heat transfer to rock from hot water, air, or steam at temperatures up to 212°F at a power level of 1-10 megawatts thermal.
- Heat transfer to rock from a typical air conditioning refrigerant.
- Cooling of typical electronic equipment.
- Heat transfer from air to drilling muck for a temperature difference between the air and muck of 10-30°F and a power level of 300 kilowatts thermal.

In each case, the heat pipes were spaced in such a manner as to be thermally non-communicating, which would be technically the most conservative, but not necessarily the most cost-effective, system. These heat pipe waste heat removal designs, while preliminary and requiring additional technical and economic optimization, did not uncover any unsurmountable barriers to implementation.

The next step in the process of validating the use of heat pipe technology for Deep Base waste heat removal requires detailed development of all of the analytical expressions and numerical analysis techniques needed to design the heat removal systems and simulate their performance. This effort: 1) includes identification of the classical analytical solutions which provide the framework and limiting cases for heat transfer in the rock, 2) development of numerical techniques for analyzing cases such as thermal interaction between heat pipes, and 3) the development of design equations which will facilitate modeling of heat pipe performance. These expressions and techniques must then be coupled together in such a manner as to allow parametric study, preferably by digital computer of the performance of various heat pipe applications for Deep Base waste heat removal. Such study will address areas of technical uncertainty, leading to the information needed to overcome design and operational problems and make reasonable system cost estimates required to determine economic feasibility.

The efforts under Phase 2 of this concept feasibility study are focused at the achievement of the technical objectives described above. The conceptual design of a Deep Base waste heat removal system utilized for this purpose is that of the heat pipe/tunnel header system conceived under Phase 1 of this study and shown in Figure 2. In this configuration, the heat pipes would be installed perpendicular to the tunnel axis and at some angle greater than horizontal, but probably less than ten degrees. Further, they would be axially staggered along the length of the header which runs along the top and bottom of the tunnel and is used to circulate the waste heat source fluid. The evaporator sections of the heat pipes would be fabricated as integral parts of the header equipment and plumbed to the heat pipes through flexible, adiabatic connectors.

Even though the conceptual design of the heat pipe/header system described above has not been optimized, and may, in fact, prove unsuitable with regard to thermal or mechanical performance or both, it is still representative of many Deep Base waste heat removal applications. The modeling expressions and techniques developed to analyze and parametrically study this system may be easily modified to take into account changes in variables such as heat pipe geometry, working fluid, or construction materials and system, configuration, or size (smaller or larger modules). For this reason, selection of the heat

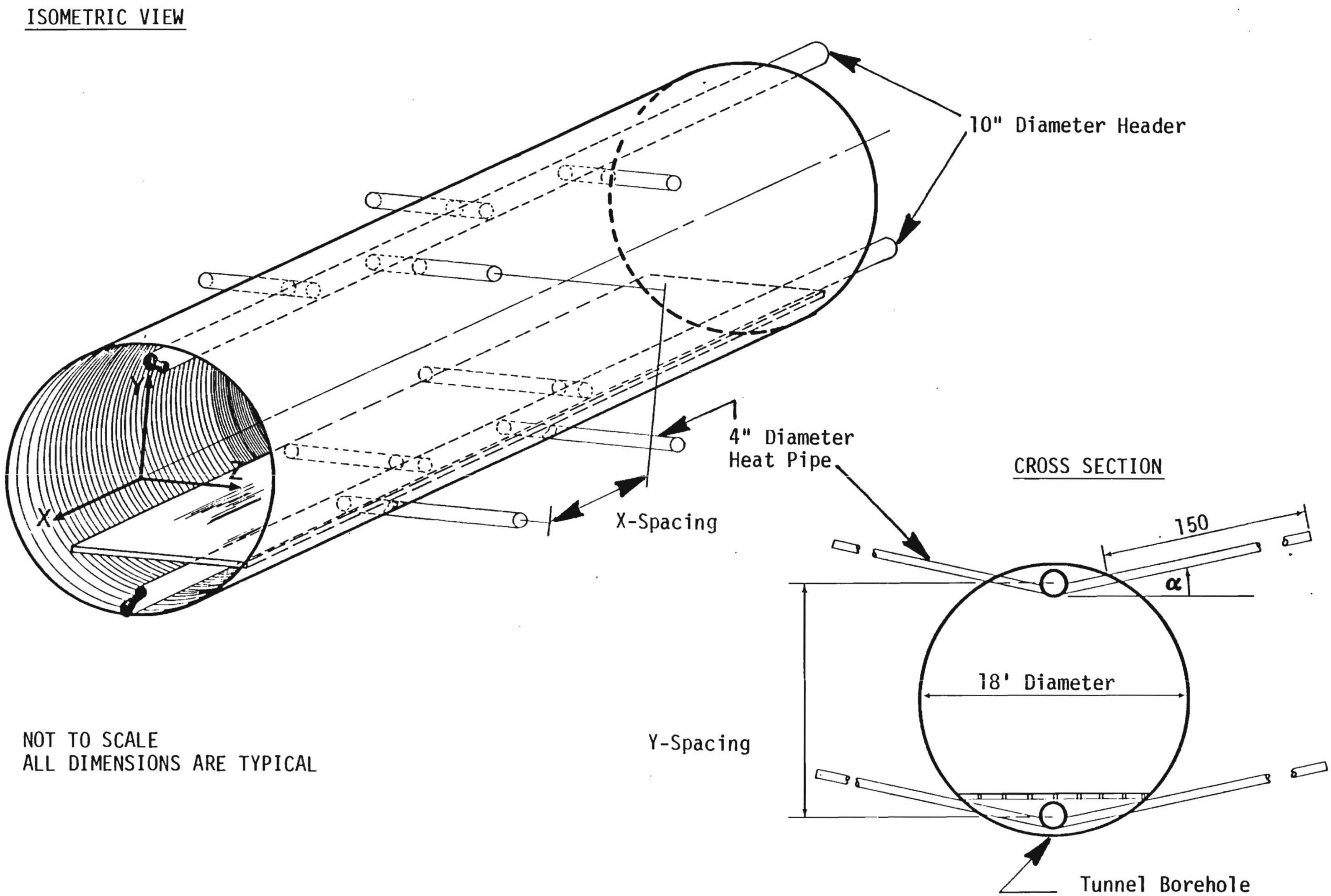


FIGURE 2

SCHEMATIC OF HEAT PIPE/TUNNEL HEADER WASTE HEAT REMOVAL SYSTEM pipe/tunnel header concept for evaluation should provide practically all of the analytical tools needed in the course of validation of thermal performance of any specific Deep Base waste heat removal application using heat pipes.

In addition to providing a good model for analyzing thermal performance of heat pipe waste heat removal applications for Deep Bases, this concept illustrates most of the mechanical performance factors such as survivability, installation, maintenance, and service life, which must be considered. The solutions found to be appropriate for mechanical performance problems associated with the heat pipe/tunnel header concept will most likely also address mechanical performance issues related to other heat pipe applications concepts. Therefore, the heat pipe/tunnel header concept provides a good overall model to facilitate study and validation of heat pipe technology for waste heat removal from Deep Bases.

II. THERMAL PERFORMANCE ANALYSIS

The following discussion details the analytical expressions needed to evaluate rock heat transfer and heat pipe performance. Also included are the results of their application to the evaluation of the heat pipe/tunnel header concept. Approximate solution techniques which are based on classical solutions to the governing differential equation, for the given geometry and sets of boundary and initial conditions are given first. Following this discussion is a development of the numerical solution techniques required to evaluate rock heat transfer cases for which the analytical expressions are inappropriate, such as thermal interaction between two or more heat pipes. Then, expressions are developed which may be used for heat pipe design or to describe their operation.

A transient model for heat pipe operation is produced by combining the expression for the instantaneous heat pipe energy balance with the rock heat transfer expressions and numerical techniques and well known equations (based on conventional heat exchanger technology) for heat transfer from the waste heat source fluid to the heat pipe evaporator. This procedure couples heat transfer characteristics of the waste heat source fluid header, heat pipes, and rock and provides the means for transient performance simulation and parametric study.

Because of the complexity and magnitude of repetitive calculations for the expressions involved, particularly those related to the numerical techniques, their use is most easily facilitated by programming them on a digital computer. Therefore, this programming step was accomplished as a prelude to all heat pipe waste heat removal system performance simulation and parametric study.

The final discussion in this report section deals with the use of the above described analytical techniques to study the heat pipe/tunnel header waste heat removal concept. The approximations to the classical, closed-form heat transfer solutions for the rock were used to provide an indication of the heat sink capability. This analysis was directed at determining the effects on heat pipe performance of rock thermal conductivity, initial rock temperature, and waste heat input, for the limiting case of thermally non-communicating heat pipes. Further, one of these solutions, the line integral expression, provides the input to the numerical analysis method used to determine the 2-D temperature profiles needed to study heat pipe spacing.

The heat pipe design equations were used to investigate effects on system performance of changes in design variables. For example, a very important parameter, because the heat pipes are gravity-assisted, is the angle of inclination.

The effect of changes in heat pipe spacing were studied through the use of 2-D temperature profiles in the rock, obtained by superposition of 1-D line integral temperature profiles. This effort provides guidance with regard to the relationships between heat pipe spacing, condenser length, and heat input into the rock.

Finally, transient operation of the heat pipe/tunnel header system is investigated. Even though the system is expected to reach nearly steady state conditions very quickly and remain there most of the time, this analysis allows one to simulate heat pipe performance during start-up, a very crucial period.

A. Analytical Expressions for Heat Transfer in the Rock

Assuming a single heat pipe transfers energy into an infinite homogenous medium (i.e., the rock) and that transport properties are constant, the governing differential equation in cylindrical coordinates for the medium is [1]:

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{1}{r^2}\frac{\partial^2 T}{\partial \Phi^2} + \frac{\partial^2 T}{\partial Z^2} = \frac{1}{\alpha}\frac{\partial T}{\partial t} \qquad \text{(A-1)}$$

where:

T = local temperature

r = radial coordinate (measured from center of heat pipe)

Φ = circumferential coordinate

Z = axial coordinate

α = thermal diffusivity

t = time

If the temperature distribution is axially symmetric ($\partial^2T/\partial\phi^2 = 0$) and if temperature distribution in the rock is independent of axial position along the heat pipe ($\partial^2T/\partial Z^2 = 0$), the governing equation becomes:

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} = \frac{1}{\alpha}\frac{\partial T}{\partial t} \tag{A-2}$$

This partial differential equation can be reduced to two ordinary differential equations which give temperature dependence on radial distance r and temperature dependence on time t, respectively.

First consider the case where rock temperature is initially uniform at T_o and the interface between rock and heat pipe is at constant temperature T_a then the general solution to Equation (2) is given by [1]:

$$T - T_o = (T_a - T_o)\left(1 + \frac{2}{\pi}\right)\int_0^{\infty} e^{-\alpha u^2 t}\left\{\frac{J_o(ur)\,Y_o(ua) - Y_o(ur)\,J_o(ua)}{J_o^2(ua) + Y_o^2(ua)}\right\}\frac{du}{u} \tag{A-3}$$

where:

J_o = Bessel function of first kind of zero order

Y_o = Bessel function of second kind of zero order

a = interface radius

On the other hand, if the rock temperature is initially uniform at T_o and heat flows through the heat pipe surface at r=a at constant rate Q, then the general solution to equation (2) is given by [1]:

$$T - T_o = -\frac{2Q}{\pi k} \int_0^{\infty} (1 - e^{-\alpha u^2 t}) \left\{ \frac{J_o(ur)\, Y_1(ua) - Y_o(ur)\, J_1(ua)}{u^2[J_1{}^2(ua) + Y_1{}^2(ua)]} \right\} du \qquad \text{(A-4)}$$

where:

J_1 = Bessel function of first kind of order one

Y_1 = Bessel function of second kind of order one

For small values of time, $\alpha t/a^2 < 0.02$, the above solutions may be simplified into more useful algebraic expressions. The small time solution for constant temperature at the heat pipe and rock interface (r=a) and uniform initial rock temperature T_o is [1]:

$$\frac{T - T_o}{T_a - T_o} = (\frac{a}{r})^{1/2} \ \mathrm{erfc}\left[\frac{r-a}{2(\alpha t)^{1/2}}\right] + \frac{(r-a)\,(\alpha t)^{1/2}}{4a^{1/2}\ r^{3/2}} \ \mathrm{ierfc}\left[\frac{r-a}{2(\alpha t)^{1/2}}\right]$$

$$+ \frac{(9a^2 - 2ar - 7r^2)\alpha t}{32\ a^{3/2}\ r^{5/2}} \ i^2\mathrm{erfc}\left[\frac{r-a}{2(\alpha t)^{1/2}}\right] + \ldots \qquad \text{(A-5)}$$

where:

$$i^n\mathrm{erfc}\ x = \int_0^{\infty} i^{n-1}\mathrm{erfc}\xi \quad d\xi,\ n = 1,2,3\ldots$$

or:

$$i^o\mathrm{erfc}\ x = \mathrm{erfc}\ x$$

$$\mathrm{ierfc}\ x = \frac{1}{p^{1/2}} \ e^{-x^2} - \mathrm{xerfcx}$$

$$i^2\mathrm{erfc}\ x = \frac{1}{4}\ [(1 + 2x^2)\mathrm{erfcx} - \frac{2}{\pi^{1/2}}\ xe^{-x^2}]$$

$$2n\, i^n \operatorname{erfc} x = i^{n-2} \operatorname{erfc} x - 2\, x\, i^{n-1} \operatorname{erfc} x$$

The small time solution for constant heat flux at the heat pipe and rock interface (r=a) and uniform initial rock temperature T_o is [1]:

$$T - T_o = \frac{2Q}{k} \left(\frac{\alpha a t}{r}\right)^{1/2} \left\{ \operatorname{ierfc} \left[\frac{r - a}{2(\alpha t)^{1/2}}\right] - \frac{(3r+a)(\alpha t)^{1/2}}{4ar} \, i^2 \operatorname{erfc} \left[\frac{r-a}{2(\alpha t)^{1/2}}\right] + \dots \right\} \tag{A-6}$$

An approximate solution for the constant heat flux case for large values of time is [1]:

$$T - T_o = \frac{Qa}{2k} \left\{ \ln \frac{4\alpha t}{cr^2} + \frac{a}{2\alpha t} \ln \frac{4\alpha t}{cr^2} + \frac{1}{4\alpha t} \left[a^2 + r^2 - 2a^2 \ln \frac{a}{r}\right] + \dots \right\} \tag{A-7}$$

where ln c = 0.5772

For relatively small pipe radii and large values of time (in general when $\alpha t/a^2 > 20$), the line integral solution is an excellent approximation for the case of constant heat input at the heat pipe and rock interface [1,2,3].

$$T - T_o = \frac{Q'}{4\alpha K} \int_{\frac{r^2}{4\alpha t}}^{\infty} \frac{e^{-u} du}{u} \tag{A-8}$$

This solution is based on an assumption that heat is added uniformly to the rock along the length of the heat pipe condenser at zero radius. Values for the line integral are available in the literature. Ingersall [2] and colleagues give comparison of line integral solutions with Carslaw and Jaeger solutions.

B. Numerical Techniques for Heat Transfer in the Rock

The previous section of this report discussed two classical solutions to the governing differential equation for heat transfer from the heat pipe into the rock. These solutions are very complex and difficult to apply but are rendered more useful by simplifying them to algebraic expressions which approximate the heat transfer in the rock for certain cases. The classical solutions give useful results only in the case of uniform initial rock temperature and either constant temperature or constant heat flux at the heat pipe and rock interface. The simpler algebraic expressions are even further restricted in that they provide good results only at certain radial distances or time values.

Unfortunately, the analytical solutions presented above, primarily because they represent one-dimensional, constant heat flux or constant temperature heat transfer to an infinite medium only, have limited use in the study of heat pipe waste heat removal systems. These expressions will be helpful for approximating heat transfer in the rock and will also provide a framework against which other models may be checked; i.e., as limiting cases. However, the study of transient operation and thermal interaction between adjacent heat pipes requires the development of additional analytical techniques based on numerical methods.

One such numerical method applicable for this study is finite difference analysis wherein energy balances are performed on very small pieces of the rock, or nodes as they are sometime called. Using the expression which represents the energy balance for a node, and well-known physical laws for heat conduction, one may develop a transient equation for its temperature. This equation may be used to calculate temperature of the node at some incremental time based on the rock thermal properties, the current temperature profile, and the quantity of heat which enters the node during the incremental time period.

The following discussion will show how this numerical method can be used to determine rock temperatures as a function of time and radial distance from the heat pipe. Appendices A.1, A.2, and A.3 give the detailed algebraic steps required to develop all of the necessary equations to apply this technique for calculating rock temperature profiles for the case at hand. For simplicity, the development is constrained at this time to consideration of one dimensional heat flow only. The most important steps from this development for both surface and interior nodes are presented here. Expressions for determination of temperature change with time for surface nodes are presented first for constant heat flux at the heat pipe and rock interface and then for variable heat flux resulting from transient heat pipe operation at the heat pipe and rock interface. Following this development, similar expressions are given for interior nodes which when combined with the surface node expressions for either the constant or variable heat flux case will result in a steady state or transient rock heat transfer model, respectively.

The constant heat flux case provides a model for steady state rock heat transfer at all times and radial distances from the heat pipe. This model may be used to check the range of applicability of the classical, analytical solutions, and is suitable for use where these solutions fail. The variable heat flux model provides the means whereby the waste heat source fluid header, the heat pipe, and the rock heat sink can be coupled together to allow study of transient operation.

As previously stated, in finite difference analysis there are two different types of nodes. The first type is the surface node which is an element on the outer surface of the system in consideration and through which heat may enter or leave. For the case being analyzed here, the surface nodes are those adjacent to the heat pipe surface. The second type of node, the interior node, is only in contact with other nodes; i.e., for this case, those that are completely surrounded by other rock elements. Figure 3 illustrates schematically a surface node and an interior node for the system under analysis here.

The following discussion presents the development of the expression for change in temperature with time of a surface node with constant heat input (see Appendix A.1 for details).

$$q_{in} = q_{out} + q_{stored} \qquad \text{(Energy Balance)}$$

or, Heat Flow into Node = Heat Flow out of Node + Heat Stored in Node

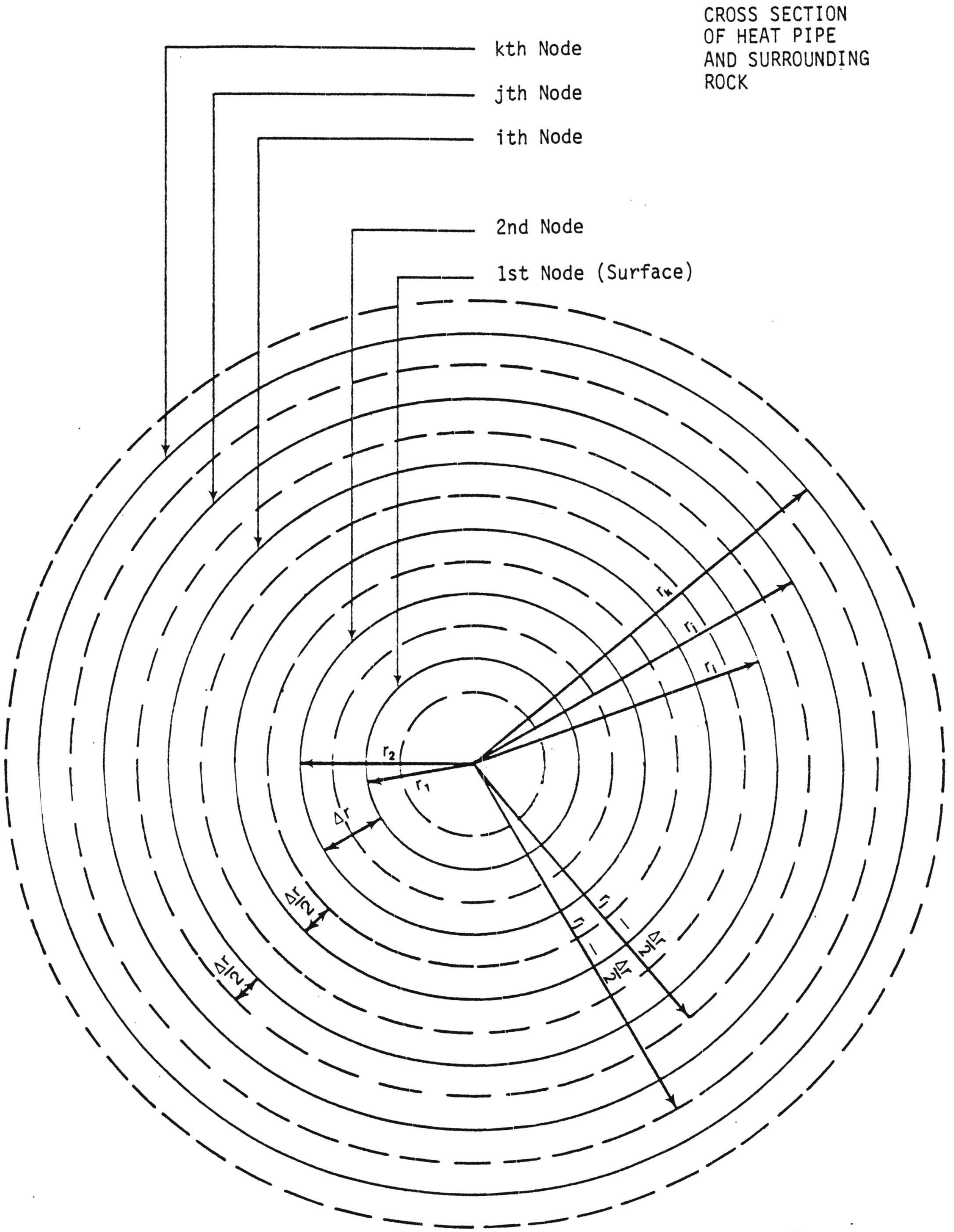


FIGURE 3

SCHEMATIC ILLUSTRATION OF SURFACE AND INTERIOR NODES

$$q_{stored} = \rho CV \frac{dT}{dt}$$

$$q_{out} = -KA \frac{dT}{dr} = 2\pi kL\ [T(2,1) - T(1,1)] / \ln (r_2/r_1)$$

$$V = \pi L[(a + \Delta r/2)^2 - a^2]$$

$$\frac{dT}{dt} = [T(1,2) - T(1,1)] / \Delta t$$

Substituting the expressions for q_{out} and q_{stored} into the energy balance and simplifying (see Appendix A.1), for the case of constant q_{in}, or constant heat flux at the heat pipe and rock interface, yields:

$$T(1,2) = T(1,1)\ \{1-[(S/B)/\ln(r_1/a)]\} + T(2,1)\ [(S/B)/\ln(r_1/a)] + 2a\Delta tQ''/(\rho CB) \tag{B-1}$$

This equation calculates the temperature T(1,2) of the rock and heat pipe interface at time 2, based on the interface temperature T(1,1) at time 1 and the assumption that q_{in}, the heat flux in, is constant. The i^{th} variable in the nomenclature for temperature T(i,n) used in Equation (B-1) describes the nodal position and while the n^{th} variable describes the time increment. For example, T(1,1) is the temperature at the first nodal (surface) position and time 1. In the same manner, T(100,2) is the temperature at the 100^{th} node (an interior node) and time 2.

In order for Equation (B-1) to be stable (i.e., valid), all of its terms must be positive. If this criterion is not satisfied, a decreasing value of temperature with time could result which is unrealistic for the system under study. Examination of Equation (B-1) shows that its second and third terms are always positive, but the first term, $T(1,1)\ [1-(S/B)/\ln(r_1/a)]$, could be negative depending on the selected nodal spacing Δr and time increment Δt. Therefore, Equation (B-1) has a single stability requirement given by:

$$T(1,1)\ [1-(S/B)/\ln(r_1/a)] \geq 0$$

or, $$(S/B)/\ln(r_1/a) \leq 1 \tag{B-2}$$

In order for Equation (B-1) to be valid, the nodal spacing and time increment cannot be chosen arbitrarily, but must be selected such that Equation (B-2) is satisfied.

The following discussion presents the development for change in temperature with time of a surface node with variable heat input supplied by the heat pipe condenser (see Appendix A.2 for details). The energy balance and expressions for q_{out} and q_{stored} are the same as for the constant heat input case presented previously. However, q_{in} is determined by the transient heat pipe energy balance equation (Equation C-36) which is developed in the next section of this report which concerns heat pipe theory and operation.

$$\dot{Q}_{out} = \frac{h_e A_e R_r}{h_e A_e R_e + R_r} [T_{in} - T_{p,c}] - \frac{C_{HP} R_r (h_e A_e R_e + 1)}{h_e A_e R_e + R_r} \frac{dT_{p,c}}{dt} = q_{in} \quad \text{(C-36)}$$

It is important to note that $\dot{Q}_{out}$ is the heat output from the heat pipe condenser which is equivalent to the heat input q_{in} at the heat pipe and rock interface. In view of this fact, the above equation may be written in the following simpler terms.

$$q_{in} = C_1 [T_{in} - T(1,1)] - C_2 [T(1,2) - T(1,1)] / \Delta t$$

Substituting the expressions for q_{in}, q_{out}, and q_{stored} into this energy balance, in a manner similar to that for the constant heat flux development, and simplifying (see Appendix A.2) will give:

$$T(1,2) = (C_1 K_1 \Delta t T_{in}) + T(1,1) [1 - C_1 K_1 \Delta t - K_1 K_2] + T(2,1) K_1 K_2 \quad \text{(B-3)}$$

The constants C_1 and C_2 are determined by the properties of the waste heat source fluid and heat pipe working fluid, respectively, and K_1 and K_2 are constants for nodes 1 and 2, respectively. Equation (B-3) couples the header, heat pipe, and rock sink and allows one to study transient operation of the waste heat removal system. This equation calculates the temperature T(1,2) of the rock and heat pipe interface at time 2, some incremental time Δt after time 1, based on the rock thermal properties and the interface temperature T(1,1) at time 1, and the heat flux q_{in} at the interface which is determined by header and heat pipe conditions.

In order for Equation (B-3) to be stable all of its terms must be positive or else, as with Equation (B-1), a decreasing value of temperature with time could occur. Examination of Equation (B-3) reveals that the first and third terms are always positive, but the second term, $T(1,1)\ [1-C_1K_1\Delta t - K_1K_2]$, could be negative depending on the selected node size Δr and time increment Δt. Therefore, Equation (B-3) has a single stability requirement given by:

$$T(1,1)\ [1 - C_1K_1\Delta t - K_1K_2] \geq 0$$

or, $$C_1K_1\Delta t + K_1K_2 \leq 1 \qquad \text{(B-4)}$$

In a manner similar to that for the previous development, the nodal spacing and time increment cannot be chosen arbitrarily, but must be selected so that Equation (B-4) is satisfied, in order to make Equation (B-3) valid.

The following discussion presents the development of the expression for change in temperature with time of an interior node; i.e., one completely surrounded by rock (see Appendix A.3 for details).

$$q_{in} = q_{out} + q_{stored} \qquad \text{(Energy Balance)}$$

$$q_{in} = 2\pi KL\ \{[T(i,1) - T(j,1)]\ /\ \ln(r_j/r_i)\}$$

$$q_{out} = 2\pi KL\ \{[T(j,1) - T(k,1)]\ /\ \ln(r_k/r_j)\}$$

$$q_{stored} = \rho\pi CL\ [(r_j + \Delta r/2)^2 - (r_j - \Delta r/2)^2]\ \frac{T(j,2) - T(j,1)}{\Delta t}$$

Substituting the expressions for q_{in}, q_{out}, and q_{stored} into the energy balance and simplifying (see Appendix A.3) yields:

$$T(j,2) = T(j,1)\ [1 - B_1 - B_2] + T(i,1)\ B_1 + T(k,1)\ B_2 \qquad \text{(B-5)}$$

where B_1 and B_2 are constants for each nodal position.

This equation calculates the temperature $T(j,2)$ of node j located at some radial distance D_j from the surface of the heat pipe, given by $D_j = (j-1)\Delta r$, at time 2. This calculation determines the temperature of node j; which is based on the temperature (at time 1) of nodes i, j, and k, the radial distance of nodes i, j, and k, and the rock thermal properties.

In order for Equation (B-5) to be stable all of its terms must be positive or else, as with Equations (B-1) and (B-3) a decreasing value of temperature with time could result. Examination of Equation (B-5) shows that its second and third terms are always positive, but the first term, $T(j,1)\ [1-B_1 - B_2]$, could be negative depending on the selected nodal spacing Δr and time increment Δt. Therefore, Equation (B-5) has a stability requirement given by:

$$T(j,1)\ [1-B_1 - B_2] \geq 0$$

or,
$$B_1 + B_2 \leq 1 \qquad \text{(B-6)}$$

In a manner similar to that for Equations (B-1) and (B-3), the nodal spacing and time increment cannot be chosen arbitrarily, but must be selected so that Equation (B-6) is satisfied, in order to make Equation (B-5) valid. Note that since a surface node equation will be used in conjunction with the interior node, Δr and Δt must be chosen to satisfy the inequalities associated with both equations.

The finite difference equations for temperature change with time derived in this section can be used to find temperature profiles in the rock heat sink in the following manner. First, the rock heat sink is to be divided into very small but finite pieces. Temperature profiles are developed by successively and repetitively applying a surface or interior node temperature equation, as appropriate, to each rock piece. Initially, the matrix of values for $T(i,1)$ at time 1 is set up with the temperature for all nodal positions being equal to the initial rock temperature. Then, starting at the surface nodes and stepping outward in increments of Δr in a radial direction, a matrix of values for $T(i,2)$ at time 2 is calculated using the finite difference temperature equations to yield a temperature profile in the rock heat sink at some incremental time Δt after time 1. Next, the temperature value for each nodal position in the matrix

T(i,1) is replaced by the temperature value from the equivalent nodal position in the matrix T(i,2); i.e., the matrix T(i,1) is set equal to the matrix T(i,2). Using this new matrix T(i,1) a new matrix T(i,2) is calculated using the finite element temperature equations to give the rock temperature profile at a still later time. This process is repeated until a time of 10,000 hours is reached which results in a complete description of rock heat sink temperature as a function of radial distance and time.

The extensive, repetitive calculations required for finite difference analysis make it an ideal candidate for programming on a digital computer. In fact, each complete temperature history developed for this study required on the order of 18 million temperature calculations. In addition, if smaller values of Δr and Δt that are selected, a more accurate temperature profile is obtained. But this increased accuracy does come at the penalty of increased computer run time and storage. Therefore, it is very important that the values for Δr and Δt be selected just small enough to obtain suitable temperature profile accuracy while minimizing the digital computer costs.

It is expected that, for the system being studied here, the heat transfer and thus, the temperature at the heat pipe and rock interface will change rapidly with time immediately following startup. However, after a few days, the heat flux and temperature change at the interface will be substantially reduced. Within a few weeks these parameters will show little change with time as steady-state operation is approached. This characteristic allows the use of a "decade" approach to maximize temperature profile accuracy while minimizing computing time and storage. This method involves the use of increasing values for nodal spacing Δr and time increment Δt for increasing "decades" of time. In other words, for the "decade" of 0.0 to 0.1 hours certain values for Δr and Δt are used, for 0.1 to 1.0 hours larger values for Δr and Δt are used, for 1.0 to 10.0 still larger values of Δr and Δt are used, and so on. On the following page, Table 1 presents the nodal spacing, time increment, and stability values associated with each "decade" which were used during the finite difference analysis performed for this study.

TABLE 1

Range of Nodal Spacing, Time Increments, and Stability Values for Finite Difference Analysis

Decade (Hours)	Δt (Hours)	Δr (feet)	Stability Criterion (B-2)		
			(B-2)	(B-4)	(B-6)
0 - 0.1	0.00025	0.00625	0.642	0.637	0.0379
0.1 - 1.0	0.001	0.0125	0.648	0.637	0.0740
1.0 - 10	0.005	0.025	0.823	0.796	0.1768
10 - 100	0.02	0.05	0.846	0.796	0.3258
100 - 1,000	0.08	0.1	0.883	0.796	0.6834
1,000 - 10,000	1/3	0.2	0.970	0.823	0.9447

NOTE: The stability values calculated using Equation (B-6) are based on a carbon steel heat pipe with water as the header and heat pipe fluids.

The finite difference analysis techniques presented here work very well for much of the heat transfer modeling required for this study. However, because the equations developed thus far in this study are for one-dimensional heat transfer to an infinite medium, they are not suitable for studying thermal interaction between adjacent heat pipes. Such study is required to determine the effects on system performance of changes in heat pipe spacing and depends on the ability to analyze two-dimensional rock heat transfer. In this case it is not realistic to assume that heat sink temperatures are a function of radial direction and time only, but, because of the influence of neighboring heat pipes, they also vary with circumferential position.

It might also be noted at this point that the assumption that axial heat transfer is negligible is probably a very good one throughout most of the length of the heat pipe condenser. However, at the end of the condenser, substantial axial heat transfer probably does occur, with the net result being an improvement in the performance of the rock heat sink. Therefore, neglecting axial heat transfer in this case is not expected to introduce serious error, but rather to provide for a more conservative analysis.

The modeling of two-dimensional heat transfer in the rock could be accomplished by finite difference analysis. The application of this numerical method would be quite complex, in this particular case, and would result in substantial requirements for computing time and storage, and thus, is outside the scope of this study. There are, however, other numerical methods which may be used to develop the desired two-dimensional rock temperature profiles. One such numerical technique involves the theory of superposition [6].

The superposition theory states simply that the temperature at any particular point is a result of all of the heat sources affecting that point. As illustrated in Figure 4, the temperature T_A at point A is equal to its initial temperature T_o plus the sum of temperature excesses Θ_i produced by neighboring heat sources. That is,

$$T_A = T_o + \Theta_1 + \Theta_2 + \Theta_3 + \Theta_4 + \ldots\ldots + \Theta_n$$

The effects of heat sources beyond a certain distance become negligible and, therefore, may be ignored.

Two-dimensional rock temperature profiles were produced by superimposing rock temperatures determined using the line integral solution. In this case, the temperature at any particular point in the rock heat sink was determined from its initial temperature and temperature excesses produced by neighboring heat pipes. It was noted that, for all practical purposes, the influence of any heat pipe at a distance of greater than about 100 feet from the point of interest was negligible and, therefore, was ignored. Details of the procedure used to develop the two-dimensional rock temperature profiles are given in Appendix A.4.

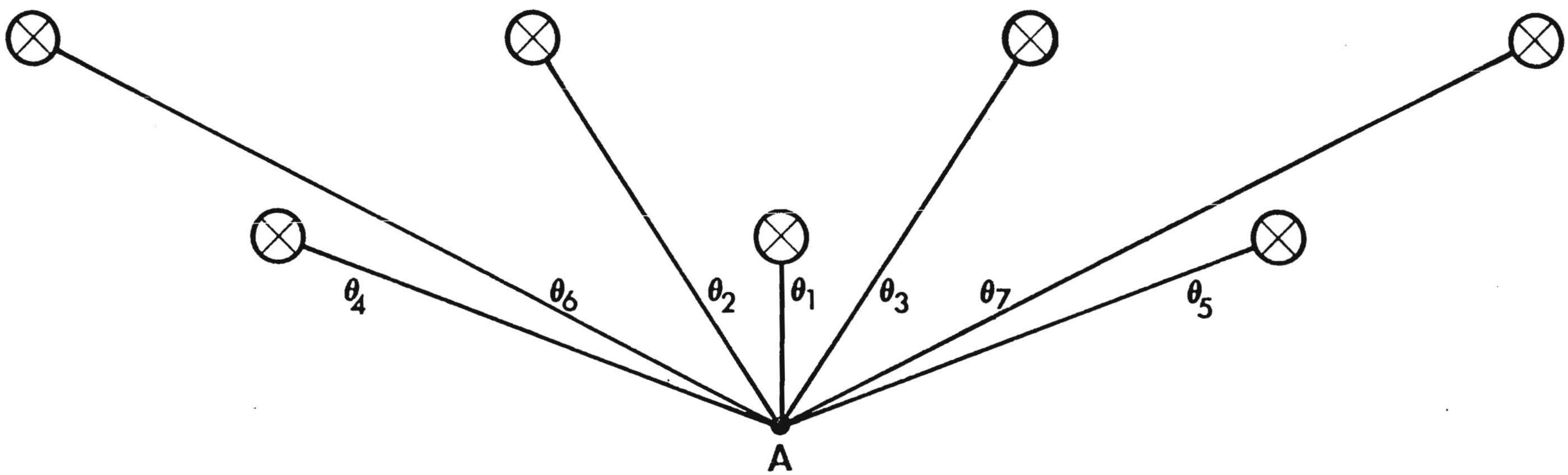


FIGURE 4

SCHEMATIC ILLUSTRATION OF SUPERPOSITION THEORY

C. Heat Pipe Theory

The primary parameters of importance in the design and operation of the heat pipe is its steady-state conductance and heat transfer limitations. Equations which describe transient operation are also useful for study of start-up and excursions from normal steady-state operation.

The equations used to describe the steady-state conductance of the heat pipe give an indication not only of its overall ability to transfer heat but also the magnitude of the thermal resistances of the various zones; i.e., evaporator wall, vapor space, capillary structure, condenser wall, etc. This information gives one the knowledge of which resistances most affect steady-state conductance and how changes in design variables can maximize this parameter.

Knowledge of the heat transfer limitations of the heat pipe determines its operational capability. Information concerning the various limitations enables one to avoid, through careful design of the individual heat pipes and overall waste heat removal systems, some potentially very serious operational problems.

The following discussion gives pertinent details of heat pipe theory related to steady-state conductance, heat transfer limitations, and transient operation. These expressions, when combined with the rock heat transfer expressions and techniques described previously, provide for overall system modeling and parametric study.

Heat Pipe Conductance. As indicated above, it is important to be able to predict on a steady basis the overall conductance of an individual heat pipe. Figure 5 is a schematic drawing showing the cross-section of a pipe and the thermal resistance of various zones in the pipe and its environment.

The resistances are:

$R_{ext,e}$ = Thermal resistance between the heat source and the exterior of the evaporator;

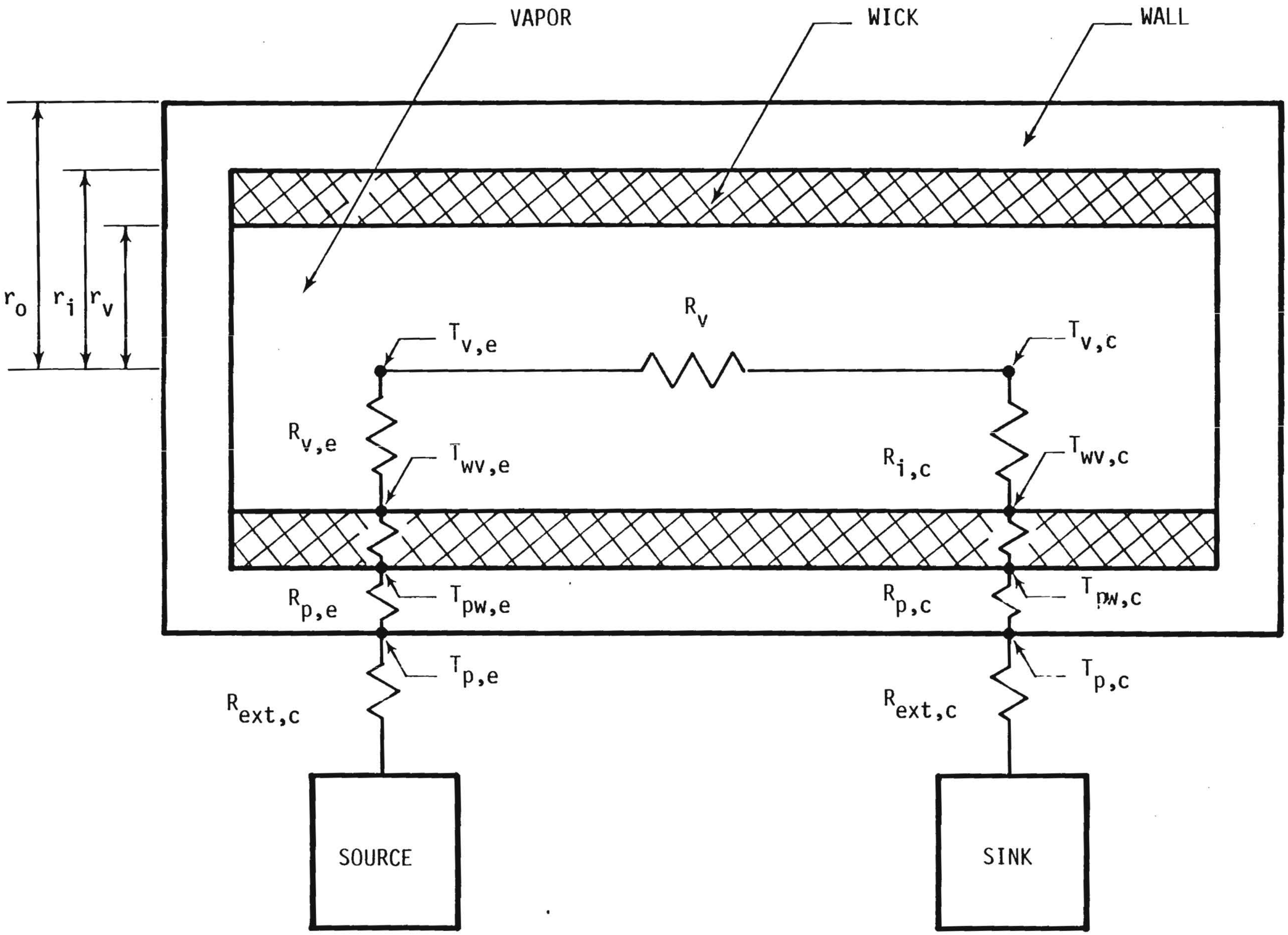


FIGURE 5

SCHEMATIC OF HEAT PIPE THERMAL RESISTANCE ZONES

$R_{p,e}$ = Thermal resistance of the pipe wall at the evaporator;

$R_{w,e}$ = Thermal resistance of the capillary structure (wick) in the evaporator;

$R_{i,e}$ = Interfacial resistance associated with the vaporization process at the evaporator;

R_v = Thermal resistance associated with the axial vapor flow (axial temperature drop);

$R_{ext,c}$ = Thermal resistance between the exterior of the condenser and the heat sink;

$R_{p,c}$ = Thermal resistance of the pipe wall at the condenser;

$R_{w,c}$ = Thermal resistance of the capillary structure in the condenser;

$R_{i,c}$ = Thermal resistance associated with the condensation process in the condenser.

The total resistance R_T including external values is:

$$R_T = R_{ext,e} + R_{p,e} + R_{w,e} + R_{i,e} + R_v + R_{i,c} + R_{w,c} + R_{p,c} + R_{ext,c} \quad \text{(C-1)}$$

Steady heat transfer is given by:

$$\dot{Q} = AU_{HP}\,(T_{p,e} - T_{p,c}) \quad \text{(C-2)}$$

where:

$\dot{Q}$ = Heat transfer rate

$T_{p,e}$ = Temperature of the pipe at the outer surface of the evaporator.

$T_{p,c}$ = Temperature of the pipe at the outer surface of the condenser.

U_{HP} = Overall heat transfer coefficient based on arbitrary area A.

Thus,

$$AU_{HP} = A_p\,U_{HP,P} = A_e\,U_{HP,e} = A_c U_{HP,c} \quad \text{(C-3)}$$

where:

A_p = Cross sectional area of pipe (based on outside diameter)

A_e = Surface area of evaporator

A_c = Surface of condenser.

and

$$U_{HP,P} = \frac{1}{\Sigma R_i A_p} = \frac{1}{\Sigma R'} \tag{C-4}$$

The thermal resistances based on the overall cross-sectional area of the pipe A_p are:

$$R'_{p,e} = \frac{r_o^2 \ln (r_o/r_i)}{2 L_e K_p} \tag{C-5}$$

L_e = evaporator length

K_p = thermal conductivity of the pipe material

$$R'_{w,e} = \frac{r_o^2 \ln (r_i/r_v)}{2 L_e K_{e,e}} \tag{C-6}$$

$K_{e,e}$ = Effective thermal conductivity of the liquid saturated wick in the evaporator end.

$$R'_{w,c} = \frac{r_o^2 \ln (r_i/r_v)}{2 L_c K_{e,c}} \tag{C-7}$$

L_c = Condenser length

$K_{e,c}$ = Effective thermal conductivity of the liquid saturated wick in the condenser end.

$$R'_{p,c} = \frac{r_o^2 \ln (r_o/r_i)}{2 L_c K_p} \tag{C-8}$$

$$R'_v = \frac{8 r_o^2 \mu_v}{v^2 r_v^2 h^2_{fg}} \left[l_a + \frac{l_e + l_c}{2} \right] \tag{C-9}$$

where:

μ_v = Vapor viscosity

l_a = Adiabatic section length

l_e = Evaporator length

l_c = Condenser length

for laminar incompressible vapor flow and,

$$R'_v = \frac{\pi \, r_o^2 \, T_v}{\rho_v \, h_{fg} \, \dot{Q}} (\Delta P_v)_{turb} \qquad \text{(C-10)}$$

$$(\Delta P_v)_{turb} = \frac{2}{r_v} \; f \; \frac{1}{2} \; \frac{V^2}{g_c} \; \left(\frac{l_e + l_c}{2} + l_a\right) = \text{Vapor pressure drop} \qquad \text{(C-11)}$$

where:

$$V = \dot{Q} \, r_u^2 \, h_{fg}$$

$$N_{Re} = \frac{VD\rho}{\mu} = \text{Reynold's Number}$$

$$f = \frac{0.0791}{N_{Re}^{1/4}}$$

for turbulent incompressible vapor flow.

The interfacial resistances, $R_{i,c}$ and $R_{i,c}$, can be predicted using kinetic theory. However, for the heat pipes under consideration in this case, both are negligible and thus not included. The overall conductance is then,

$$U_{HP,P} = \frac{1}{R'_{P,e} + R'_{w,e} + R'_v + R'_{w,c} + R'_{P,c}} \qquad \text{(C-12)}$$

or,

$$U_{HP,e} = \frac{U_{HP,P} \, A_P}{A_e} \qquad \text{(C-13)}$$

The equations listed above assume no overfill of liquid. Since gravity is the main pumping mechanism in the present case, some liquid will be in the bottom of the pipe as shown in Figure 6, and it will be necessary to include this in thermal resistance equations. Therefore, taking into account fluid overfill, the thermal resistance equations are given by:

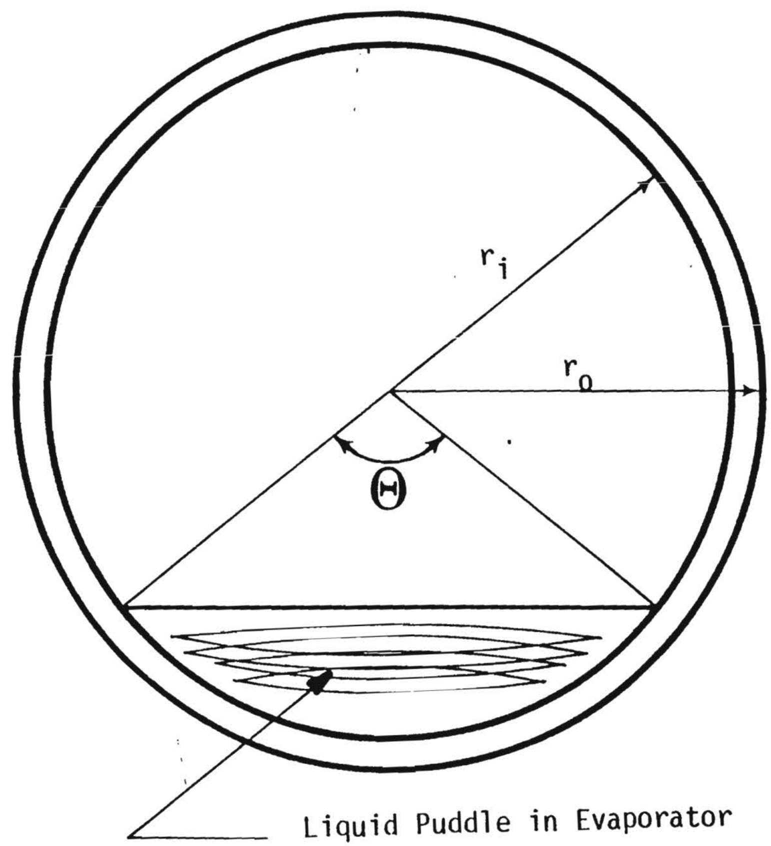


FIGURE 6

CROSS SECTION OF HEAT PIPE EVAPORATOR SHOWING PUDDLE ANGLE

$$R_{p,e} = \frac{\ln (r_o/r_i)}{(2 - \Theta)\ L_e K_p} \qquad \text{(C-14)}$$

$$R_{w,e} = \frac{\ln (r_i/r_o)}{(2 - \Theta)\ L_e K_{e,e}} \qquad \text{(C-15)}$$

$$R_{p,c} = \frac{\ln (r_o/r_i)}{2\ \pi\ L_e K_p} \qquad \text{(C-16)}$$

$$R_{w,c} = \frac{\ln (r_i/r_o)}{2\ \pi\ L_c K_{e,c}} \qquad \text{(C-17)}$$

and, the overall conductance is then,

$$U_{HP,P} = \frac{1}{(R_{P,e} + R_{w,e} + R_v + R_{w,c} + R_{P,c})\ A_p} \qquad \text{(C-18)}$$

or

$$R'_i = R_i A_p \quad \text{where} \quad A_p = \pi r_o^2 \qquad \text{(C-19)}$$

also, the overall heat transfer coefficient for the evaporator may be found by:

$$U_{HP,e} = \frac{U_{HP,p}\ A_p}{A_e} \qquad \text{(C-20)}$$

where:

$$A_e = L_e\ r_o\ (2\pi - \alpha)$$

The vapor resistance for laminar incompressible flow is given by:

$$R_v = \frac{8\ \mu_v\ T_v}{\pi\ v^2\ r_v^4\ h^2{}_{fg}\ g_c} \left[l_a + \frac{l_c}{2} + \frac{l_e}{(2\pi - \alpha + \sin \Theta)} \right] \qquad \text{(C-21)}$$

Wick porosity for a tightly wrapped screen is:

$$\varepsilon = 1 - \frac{\pi\ SND}{4} \qquad \text{(C-22)}$$

where:

S = Crimping factor ≅ 1.05

N = Mesh number

D = wire diameter

Wick permeability may be estimated as,

$$K = \frac{d^2 \; \varepsilon^3}{122 \; (1 - \varepsilon)^2} \qquad \text{(C-23)}$$

where:

D = wire diameter

ε = wick porosity

An effective thermal conductivity for a single layer of screen saturated with a liquid can be estimated as,

$$K_1 = \frac{K_1 \; [(K_1 + K_w) - (1 - \varepsilon) \; (K_1 - K_w)]}{[(K_1 + K_w) + (1 - \varepsilon) \; (K_1 - K_w)]} \qquad \text{(C-24)}$$

where:

K_1 = Thermal conductivity of working fluid

K_w = Thermal conductivity of the screen wick material

ε = Wick porosity

Heat Transfer Limitations Within the Heat Pipe. Though conceptually a simple device, the heat pipe is very complex from an operational standpoint. The output from a heat pipe can be limited by a number of considerations: if the velocity of the vapor reaches sonic velocity, "choking" will occur; entrainment of liquid by the vapor must be avoided, or the evaporator could be starved of liquid; film boiling must be avoided since this results in poor heat transfer coefficients; and finally, the rate of circulation of fluid obtainable is limited by the available pumping force (capillary pressure or gravity flow) [7]. Figure 7 illustrates diagrammatically how these four factors combine to give the operational envelopes for a given design of heat pipe. At low vapor pressures the sonic velocity may be the limiting factor as the gas density will be low; this is the region 1-2 in Figure 7. Entrainment limits the heat flow in

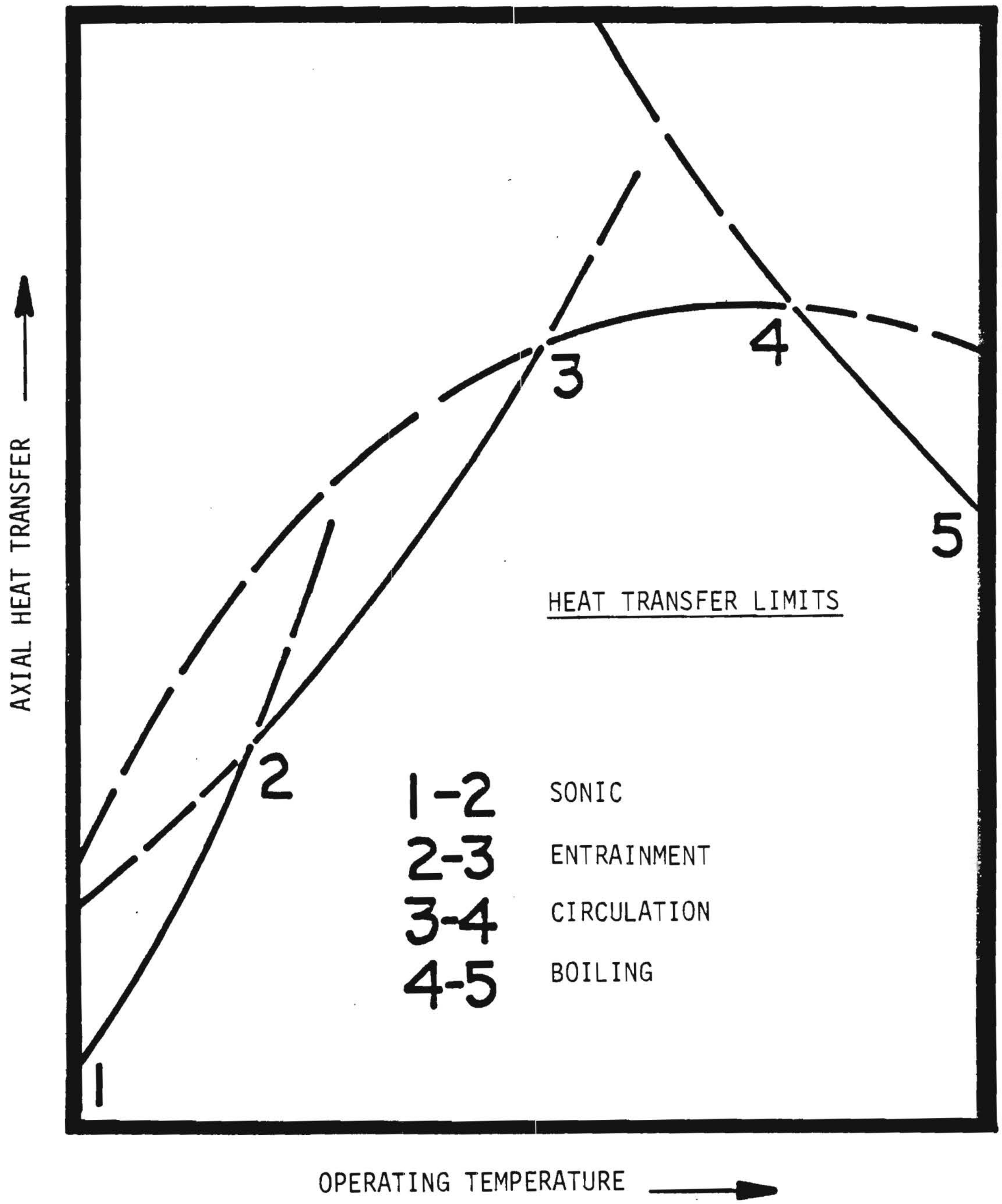


FIGURE 7

HEAT PIPE OPERATIONAL ENVELOPE: MAXIMUM HEAT TRANSFER CAPACITY AS A FUNCTION OF OPERATING TEMPERATURE region 2-3. In region 3-4, the restriction on output is due to limitations of the available pumping force. In region 4-5 the axial heat flow is limited by the onset of film boiling.

The primary heat transfer limitation in the present application is the maximum flow of liquid which can occur from the condenser zone to the evaporator zone. This maximum flow will depend on the angle of inclination of the pipe, the pipe geometry, and the viscosity of the working fluid in the liquid state.

Assume a two-dimensional constant depth liquid stream flowing down an incline with negligible shear at the vapor liquid interface, as shown in Figure 8.

The velocity variation with y is given by

$$V_x(y) = \frac{\rho g_c}{\mu}\,[hy - y^2/2]\,\sin\alpha \qquad \text{(C-25)}$$

Integrating and assuming a width of b for the stream, the maximum mass flow rate is given by:

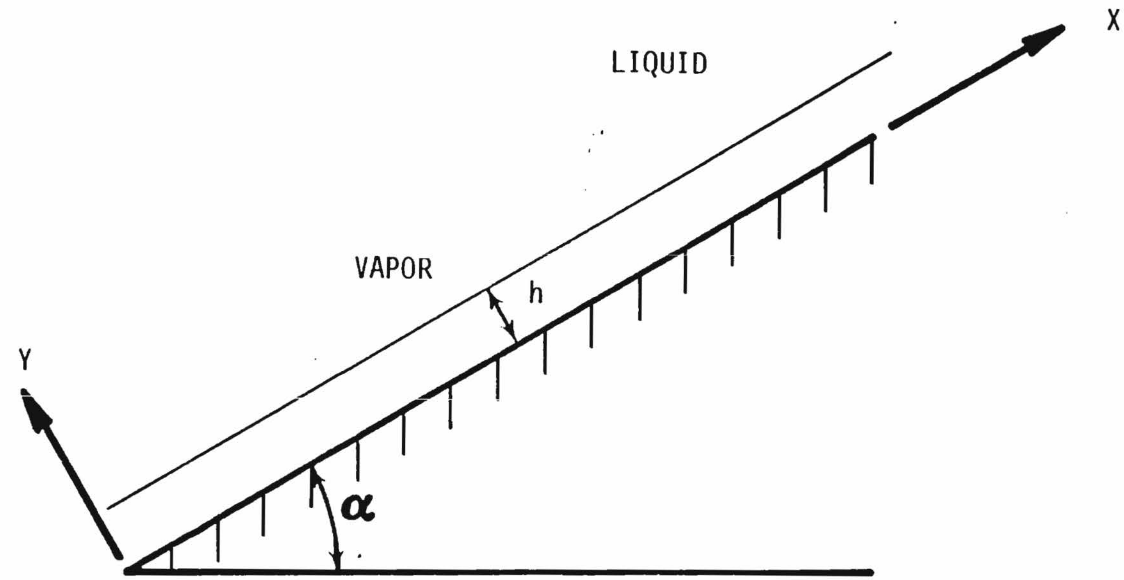
$$\dot{M} = \frac{b\rho^2 g h^3}{3\mu}\,\sin\alpha \qquad \text{(C-26)}$$

and the associated maximum heat transfer will be,

$$\dot{Q}_{g,max} = \frac{b\rho^2 g_e h^3 h_{fg}}{3\mu}\,\sin\alpha \qquad \text{(C-27)}$$

where h_{fg} is the latent heat of vaporization and $\dot{Q}_{g,max}$ is the gravity limitation.

Approximately, $b = 2r_v \sin(\Theta/2)$, where Θ is the puddle angle as shown in Figure 6.



Angle of Inclination is Exaggerated
Liquid Height h varies

FIGURE 8

MODEL FOR LIQUID FLOW IN HEAT PIPE CONDENSER

To account for variation in height h along the length of the condenser and adiabatic sections, an average value of h may be used. Assuming a linear variation of liquid height through condenser and adiabatic section, the volume V of free liquid (that is not contained in the pores of the capillary structure) is given by:

$$V = \frac{r_v^2}{4} (1 - \sin \Theta_1)(L_a + L_c) \qquad \text{(C-28)}$$

When no liquid stream is present, the sonic limitation will be,

$$\dot{Q}_{s,max} = A_v \rho_o h_{fg} \left[\frac{\gamma_v R_v T_o}{2 (\gamma_v + 1)}\right] \qquad \text{(C-29)}$$

$\dot{Q}_{s,max}$ = sonic limitation

A_v = Vapor core cross sectional area

ρ_o = Stagnation density of the vapor

h_{fg} = Latent heat of vaporization

γ_v = Specific heat ratio

R_v = Gas constant of a particular gas

T_o = Stagnation temperature of vapor (upstream end of evaporator section)

Accounting for the flowing stream of liquid gives for the sonic limitation,

$$\dot{Q}_{s,max} = \rho_o \frac{r_v^2}{2} (2\pi - \Theta + \sin \Theta)\, h_{fg} \left[\frac{\gamma_v R_v T_o}{2 (\gamma_v + 1)}\right]^{1/2} \qquad \text{(C-30)}$$

An entrainment limit, neglecting puddling in the evaporator, is given by:

$$\dot{Q}_{e,max} = A_v\, h_{fg} \left(\frac{\sigma \rho_v g_c}{2 r_{h,s}}\right)^{1/2} \qquad \text{(C-31)}$$

$\dot{Q}_{e,max}$ = entrainment limitation

Av = vapor core cross-sectional area

σ = surface tension

ρ_v = vapor density

$r_{h,s}$ = hydraulic radius of the wick surface pores

(for screen wicks $r_{h,s}$ = 1/2 wire spacing and $2r_{h,s} \cong$ pore opening)

When the flowing liquid stream is accounted for the entrainment limit becomes:

$$\dot{Q}_{e,max} = \frac{r_v^2}{2} (2\pi - \Theta + \sin\Theta)\, h_{fg} \left[\frac{\sigma\, \rho_v\, g_c}{2\, r_{h,s}}\right]^{1/2} \tag{C-32}$$

The boiling limit for a conventional heat pipe without overfill is given by:

$$\dot{Q}_{b,max} = \frac{2\pi\, L_e\, K_e\, T_v}{h_{fg}\, \rho_v\, \ln(r_i/r_v)} (2\theta/r_n) \tag{C-33}$$

$\dot{Q}_{b,max}$ = boiling limitation

L_e = evaporator length

K_e = effective thermal conductivity of the liquid saturated wick

T_v = vapor temperature

r_i = inner radius of the pipe container

r_v = vapor core radius

h_{fg} = latent heat of vaporization at T_v

ρ_v = vapor density at T_v

σ = surface tension

r_n = nucleation radius of the vapor bubbles

Considering overfill

$$Q_{b,max} = \frac{(2\pi - \alpha)\, L_e\, K_e\, T_v}{h_{fg}\, \rho_v\, \ln(r_i/r_v)} (2\sigma/r_n) \tag{C-34}$$

Transient Heat Pipe Operation. Assuming that the heat pipe operates at all times at heat transfer rates below limiting values, temperature variations in the vapor space with respect to position will be relatively small at an instant of time. One may write;

$$Q_{in} - Q_{out} = C_{HP} \frac{dT}{dt} \tag{C-35}$$

where:

Q_{in} = Heat transfer into the heat pipe at the evaporator surface

Q_{out} = Heat transfer into the rock

C_{HP} = Heat capacity of the heat pipe

$\frac{dT}{dt}$ = Rate of change of temperature of heat pipe with time

The temperature T is a mean temperature for the pipe at an instant of time and it taken equal to the vapor temperature within the pipe. If energy is transferred to the outside of the heat pipe by a fluid such as in a header with constant temperature T_{in} and film coefficient h_e, then:

$$Q_{out} = \left(\frac{h_e A_e R_r}{h_e A_e R_e + R_r}\right) (T_{in} - T_{p,c}) - \frac{C_{HP} R_r (h_e A_e R_e + 1)}{(h A_e R_e + R_r)} \frac{dT_{p,c}}{dt} \tag{C-36}$$

where:

h_e = Heat transfer coefficient on outside evaporator surface

A_e = Outside evaporator area

R_e = $R_{p,e} + R_{w,e}$ = Total thermal resistance in heat pipe at evaporator

R_T = $R_{p,e} + R_{w,e} + R_{p,c} + R_{w,c}$ = Total heat pipe thermal resistance

R_r = R_e/R_T

T_{in} = Temperature entering evaporator external heating zone

$T_{p,c}$ = Temperature on outside of heat pipe condenser

C_{HP} = Total heat capacity of heat pipe

t = Time

This equation may be used to couple characteristics of header, heat pipe, and rock since $T_{p,c}$ is the temperature at the heat pipe and rock interface and Q_{out} is the heat transfer to the rock.

D. Analysis of Heat Pipe/Tunnel Header System

As discussed in the introduction of this report, the heat pipe/tunnel header concept will provide a good initial model for analyzing the thermal performance of heat pipe technology for waste heat removal in Deep Bases. This concept is illustrated by Figure 2 and described in the Introduction. Typical heat pipe dimensions, which have not been optimized, are selected to be four inches for the outer diameter and 150 feet for condenser length. The average operating temperature for the heat pipes based on anticipated Deep Base waste heat sources, is estimated to be on the order of 200°F. Either water or methanol could be chosen for the heat pipe working fluid, since both are suitable at this operating temperature.

This section of the report details the results of thermal performance analysis of the heat pipe/tunnel header concept. The evaluation involves use of the analytical expressions and numerical techniques developed previously to study a number of important thermal performance issues. These issues include: (1) effect of the physical properties of the rock environment such as thermal conductivity and initial temperature; (2) evaluation heat pipe design variables such as length, evaporator heat load, and angle of inclination; (3) effect of waste heat removal system design variables such as heat pipe spacing; and (4) transient response of the waste heat removal system.

Considerable investigation of the literature was focused at determination of representative rock physical properties to be used in thermal performance analyses [13,14,15]. Lack of specific information regarding the expected location of the Deep Base resulted in the selection of a range of physical properties to be considered. These ranges were presented in the Phase 1 report for this study and are listed again in Table 2 and are based on the anticipated geology of various potential Deep Base locations with the average value listed being that which was given most often in the literature for granite. The range of values for rock thermal conductivity provide a best (K = 2.3 BTU/hr-ft-F), average (K = 1.6 BTU/hr-ft-F), and worst case (K = 1.0 BTU/hr-ft-F) for consideration in the thermal performance analyses.

TEMPERATURE PROFILE

INFINITE MEDIUM AND 10,000 HOURS

(Line Integral)

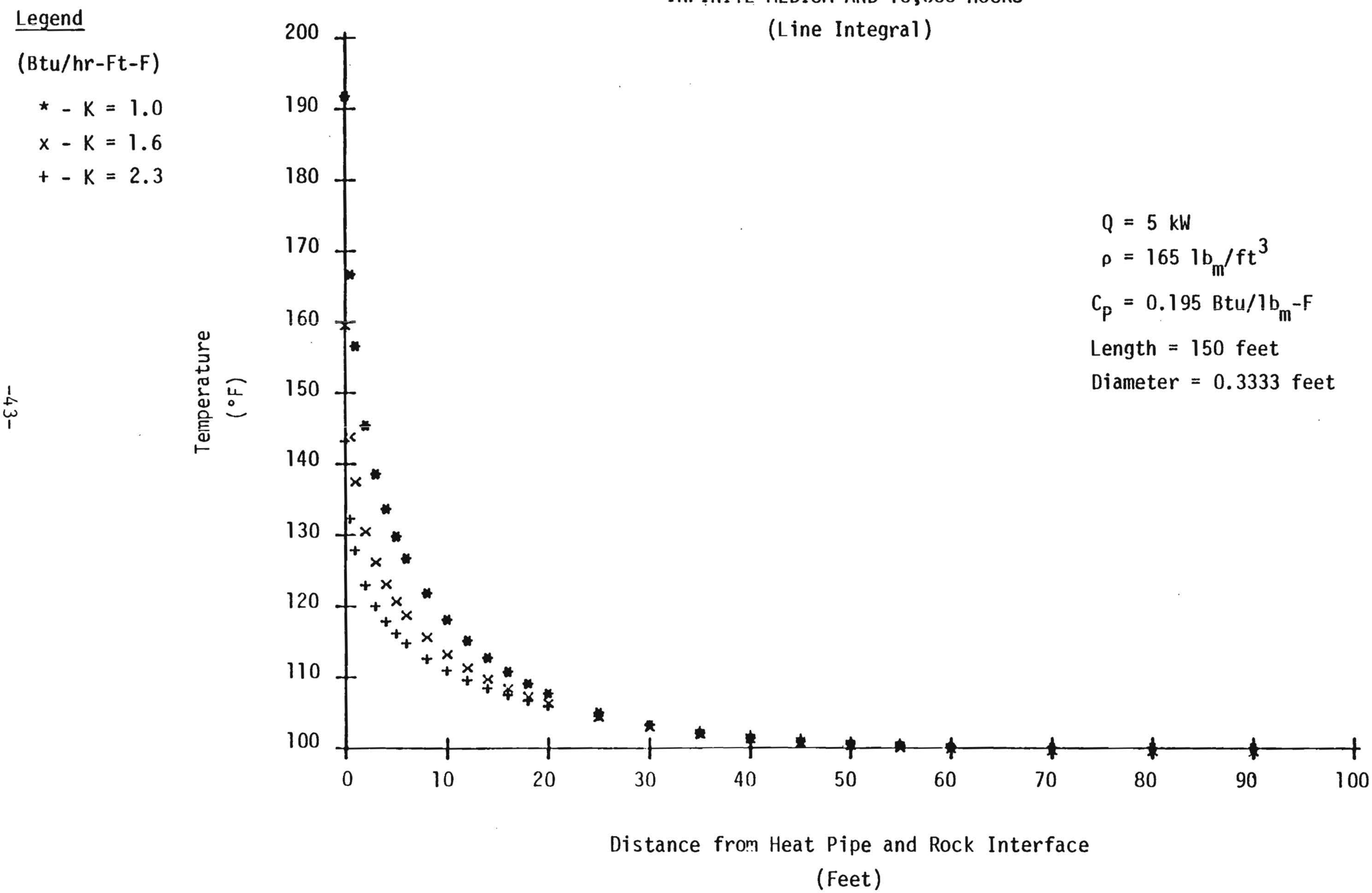


FIGURE 9

EFFECT OF ROCK THERMAL CONDUCTIVITY ON ROCK TEMPERATURE PROFILES

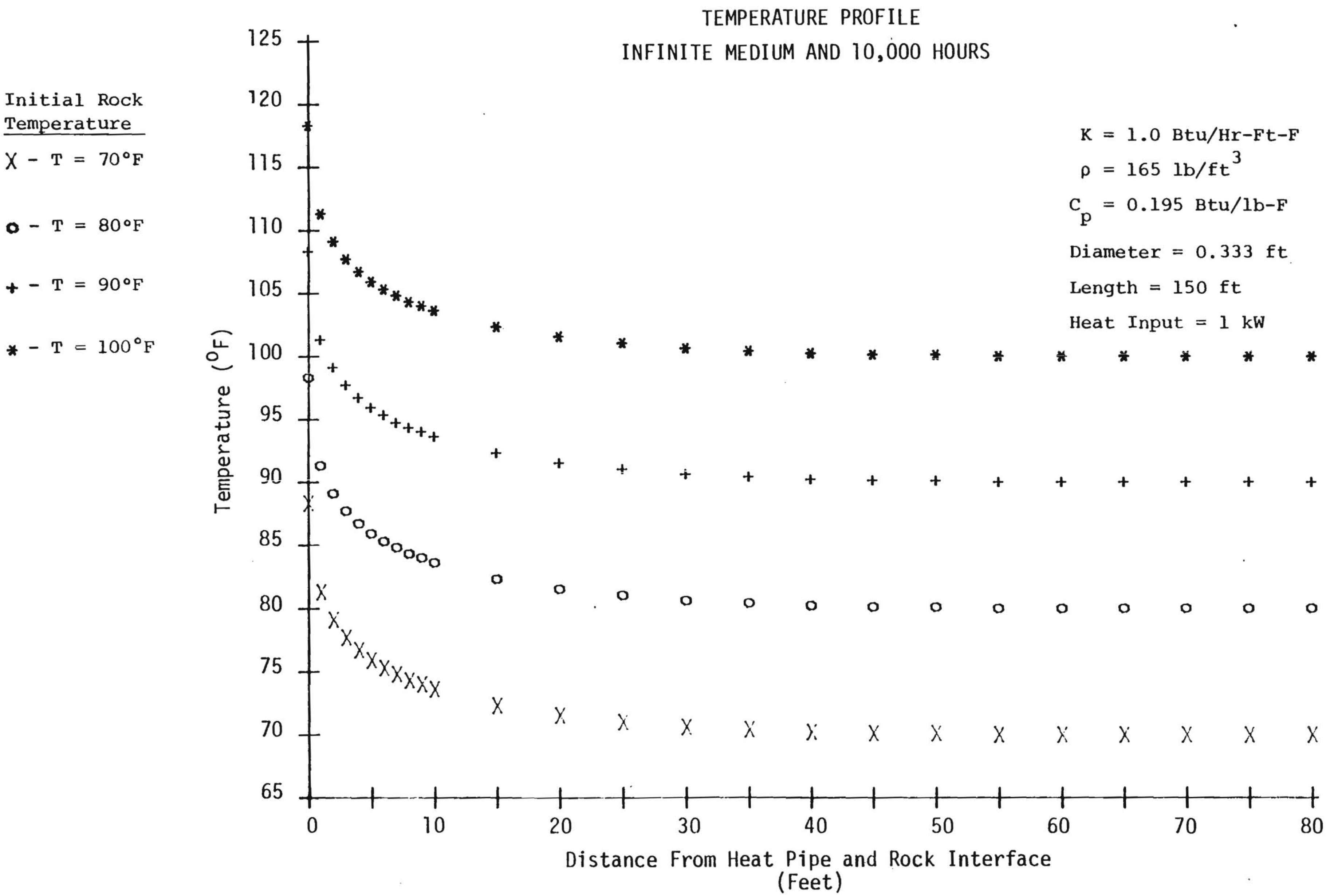


FIGURE 10

EFFECT OF INITIAL ROCK TEMPERATURE ON ROCK TEMPERATURE PROFILES each heat pipe has a constant heat output of 5 KW and is transferring heat to an infinite amount rock.

The effect of initial rock temperature is pronounced resulting in a difference of approximately 30°F in the interface temperature if the difference in initial rock temperature is this same amount. As in Figure 9, the slope of the profiles changes very little at a point in the range of 15-20 feet from the heat pipe and rock interface so that if the heat pipes are spaced at this distance or greater they can be assumed to be effectively non-communicating. In addition, it may be noted that, if the Deep Bases were sited in rock with low thermal conductivity, the reduced thermal performance caused by poor heat transfer in the rock could be offset somewhat if the rock had a lower initial temperature.

Figure 11 was also generated using the line integral solution and it illustrates the effect of varying the linear thermal output of each heat pipe. The profiles shown are at time equal to 10,000 hours and a rock thermal conductivity of 1.6 BTU/hr-ft-F for heat transfer to an infinite amount of rock.

Variation in linear heat output for individual heat pipes may be accomplished by two different means; 1) changes in heat input to the evaporator for a heat pipe of given length and diameter (1, 3 and 5 KW/heat pipe with a condenser length and diameter of 150 feet and 4 inches, respectively) and 2) changes in heat pipe condenser length for a constant diameter heat pipe with constant evaporator heat input. The temperature profiles in Figure 11 indicate virtually the same rock heat transfer characteristics as seen in the previous two figures. Furthermore, an order of magnitude increase in linear heat output, from 0.0067 KW/ft to 0.0333 KW/ft, results in a significant difference of more than 40°F for the temperature at the heat pipe and rock interface after 10,000 hours. The effect of varying condenser diameter could be studied in a similar manner.

As discussed in the previous section concerning heat pipe theory, there are four significant limitations to heat transfer for a heat pipe; vapor sonic flow, liquid entrainment, film boiling, and rate of fluid circulation. For the size heat pipes and operating conditions considered so far in this study, the first

TEMPERATURE PROFILE

INFINITE MEDIUM AND 10,000 HOURS

(Line Integral)

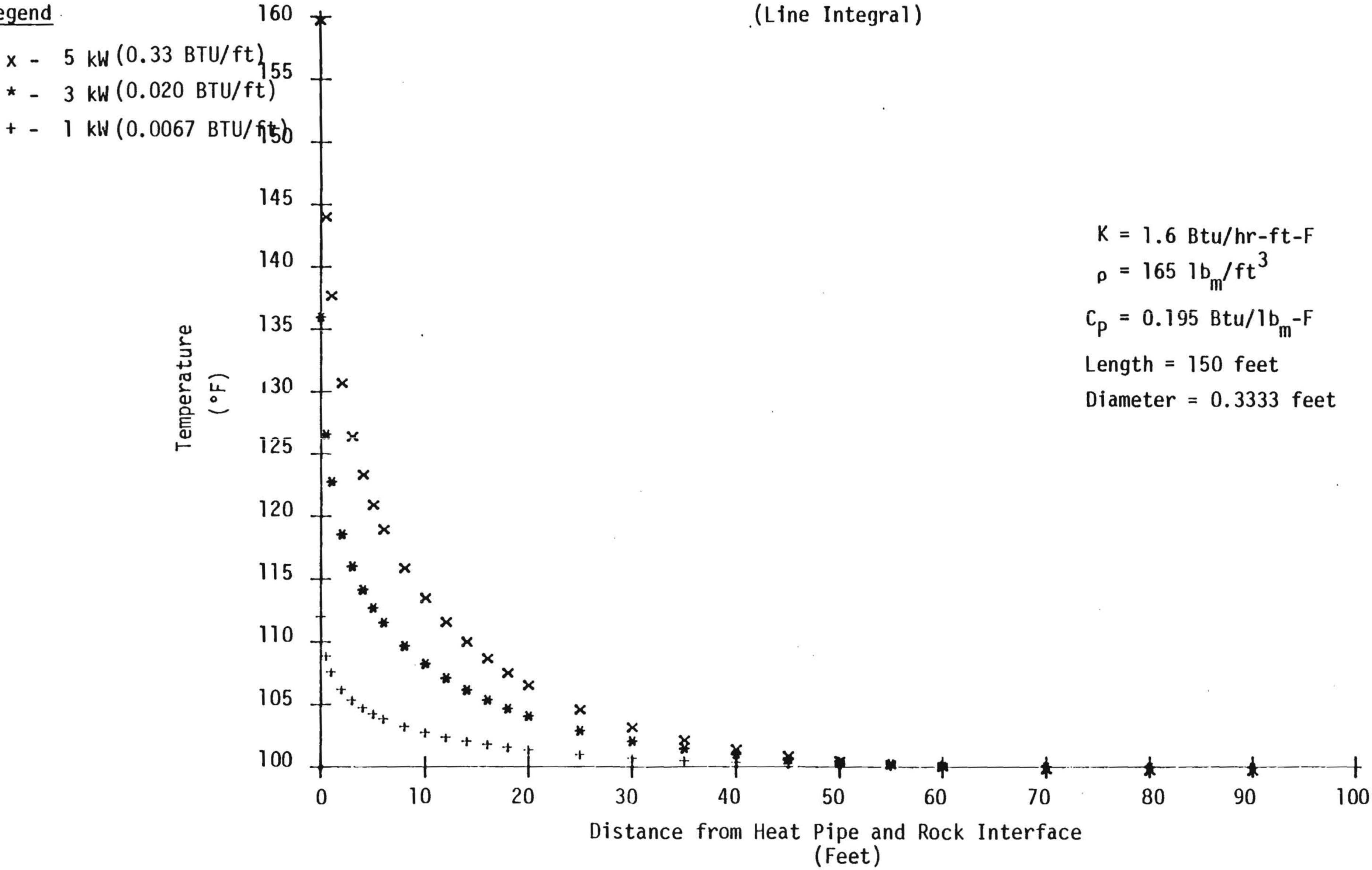


FIGURE 11

EFFECT OF HEAT PIPE LINEAR THERMAL OUTPUT ON ROCK TEMPERATURE PROFILES three limitations pose no problems, since they are all expected to be well above maximum heat transfer rates. Values for these limitations are highly dependent on heat pipe design and operating variables (as an example, some typical numbers are shown in Table 3). The fluid circulation limitation is of major importance, since it is determined by gravity flow which is related to the angle of inclination above horizontal for the heat pipe.

TABLE 3

TYPICAL VALUES FOR HEAT PIPE HEAT TRANSFER LIMITATIONS*

Heat Transfer Limitations	$\dot{Q}$, Million BTU/hr
Sonic	15.3
Entrainment	0.62
Boiling	0.69

* Carbon steel heat pipe with water as the working fluid at an average operating temperature of 212°F. The maximum heat pipe load for a carbon steel system would be 57,000 BTU/hr.

Figure 12 is a plot of angle of inclination as a function of condenser length, with increasing thermal loads, for a heat pipe of four inches outer diameter operating at 200°F with water as the working fluid overfilled by 50 percent (i.e., the heat pipe contains 50 percent more working fluid than that required to completely saturate the capillary structure). Obviously, the longer the condenser, the greater the required angle of inclination to overcome the viscous forces opposing the gravity flow return of liquid to the evaporator.

The angle of inclination remains relatively low for all heat pipe thermal loadings illustrated in Figure 12 for condenser lengths up to 220 feet, but

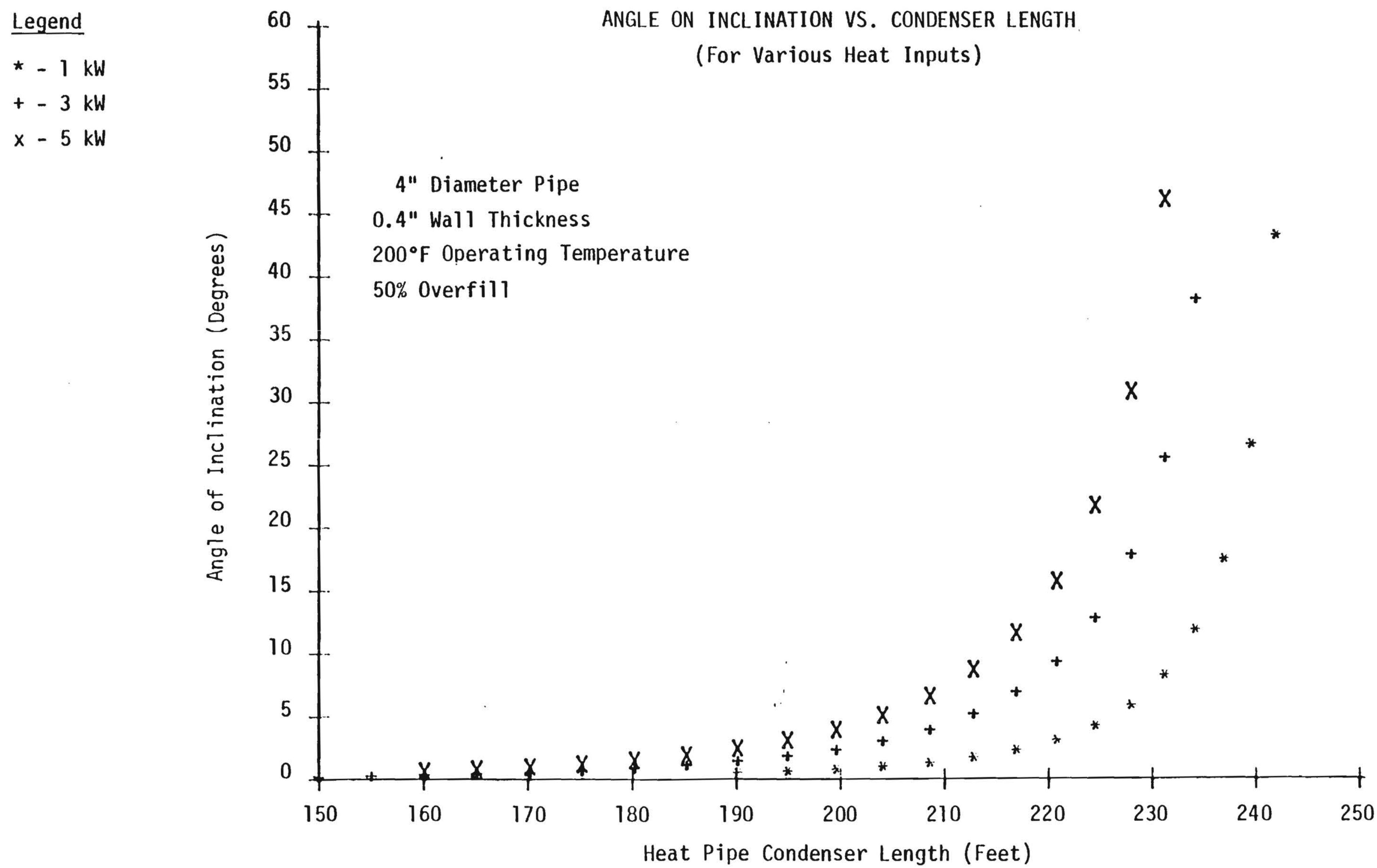


FIGURE 12

HEAT PIPE ANGLE OF INCLINATION AS A FUNCTION OF CONDENSER LENGTH

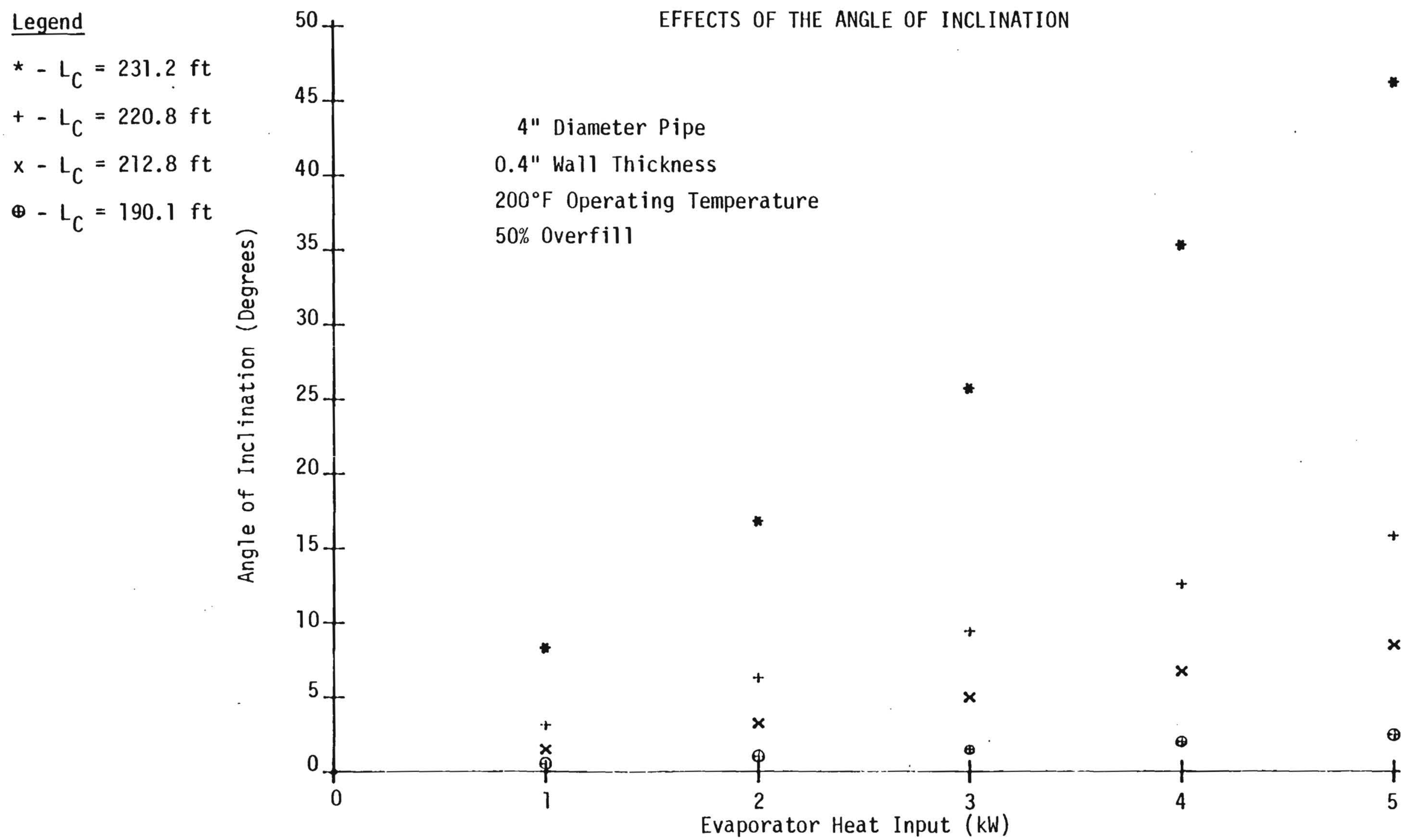


FIGURE 13

ANGLE OF INCLINATION AS A FUNCTION OF EVAPORATOR HEAT INPUT

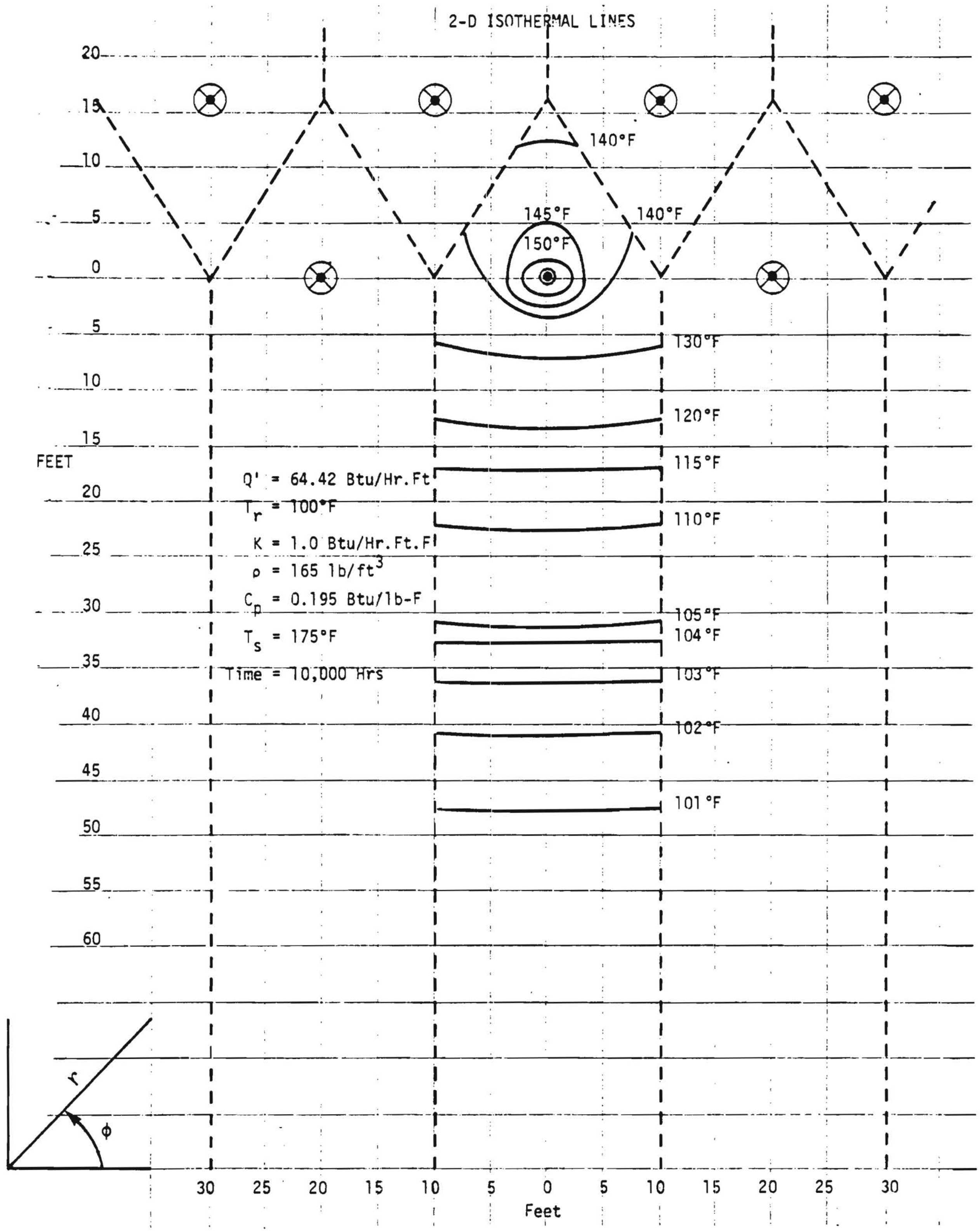


FIGURE 14

TWO-DIMENSIONAL ROCK TEMPERATURE PROFILES

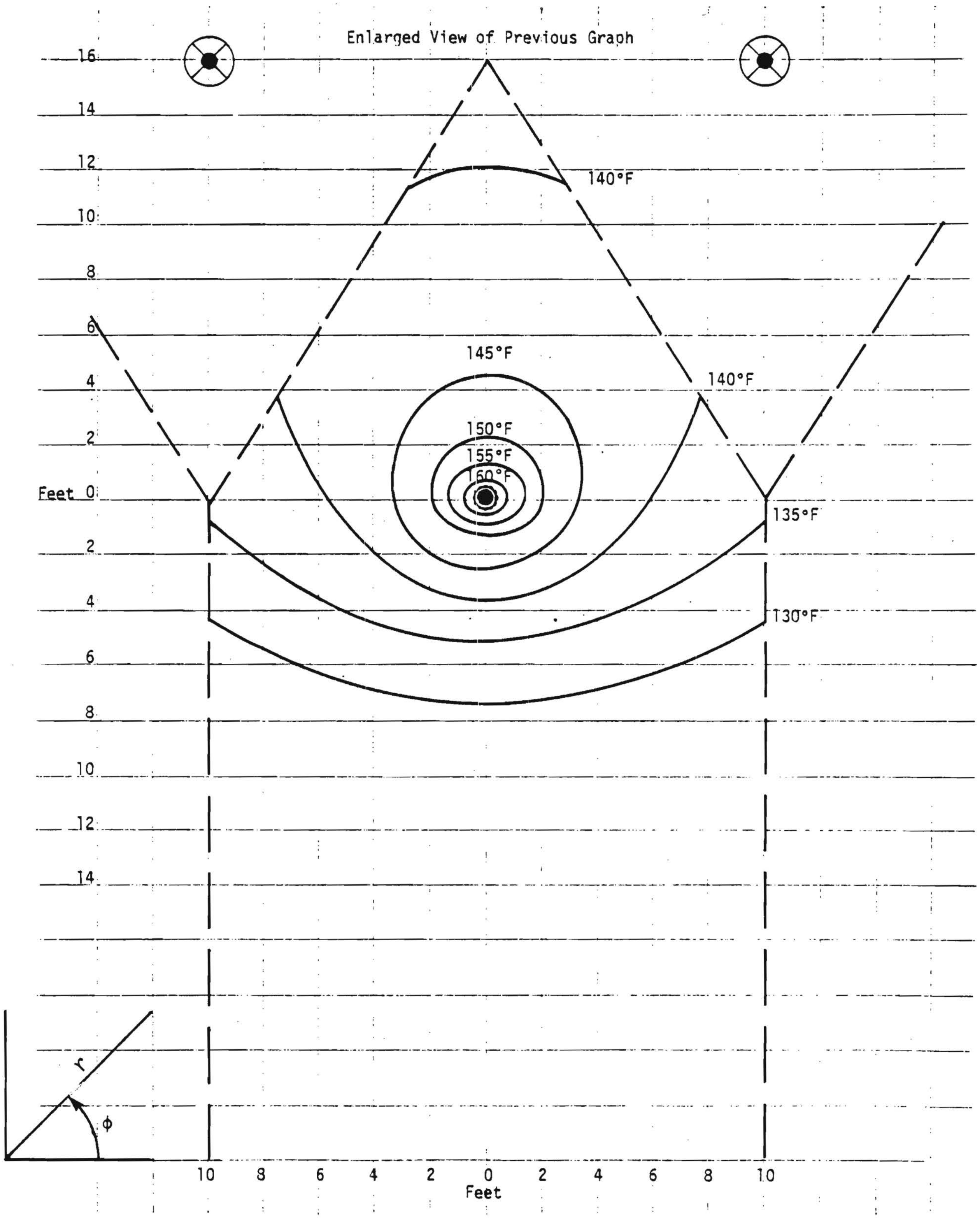


FIGURE 15

TWO-DIMENSIONAL TEMPERATURE PROFILES NEAR HEAT PIPE

Legend

+ - K = 1.0 Btu/hr-ft-F

* - K = 1.6 Btu/hr-ft-F

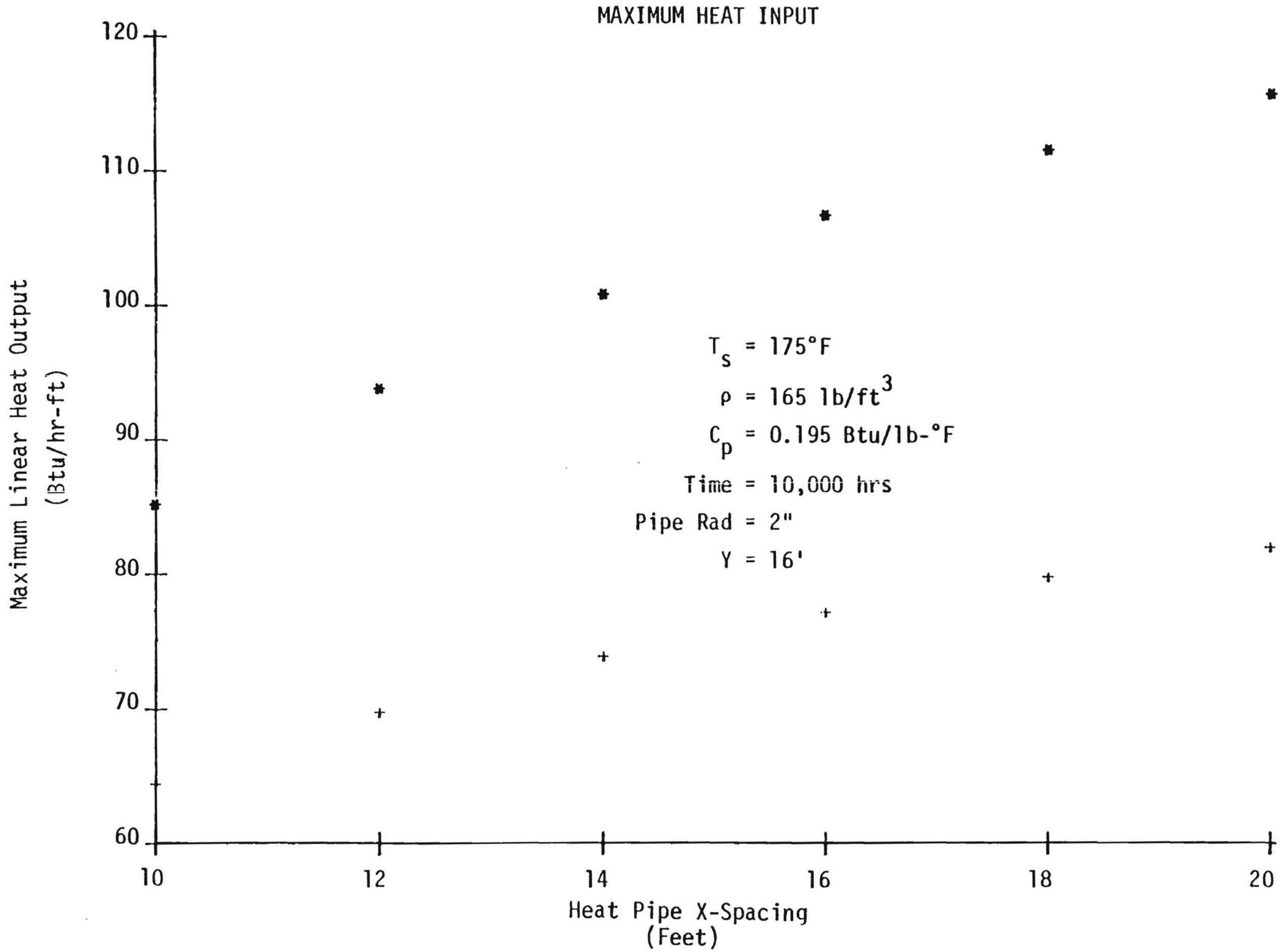


FIGURE 16

MAXIMUM LINEAR HEAT OUTPUT AS A FUNCTION OF HEAT PIPE X-SPACING of 16 feet. The data is plotted for two different rock thermal conductivities, low (K = 1.0 BTU/hr-ft-F) and average (K = 1.6 BTU/hr-ft-F), and assumes the initial rock temperature is 100 °F and the temperature at the heat pipe and rock interface is limited to 175 °F.

A value of Q' in Figure 16 corresponding to a particular X-spacing is the maximum linear heat pipe thermal loading possible at that X-spacing if the interface temperature is limited to 175 °F after time equal to 10,000 hours. For example, if the rock thermal conductivity is 1.0 BTU/hr-ft-F and the X-spacing is 16 feet, the maximum linear heat output is 77.2 BTU/hr-ft. Therefore, if the heat pipe was required to dissipate a constant 5 KW of heat over a 10,000 hour time period, the condenser length required would be approximately 221 feet (condenser length is given by total heat output divided by linear heat output, (5 KW x [3413 BTU/hr-KW]/[77.3 BTU/hr-ft] = 221 ft). A graph such as Figure 16 can, of course, be generated for any given set of conditions; i.e., various rock properties, y-spacings, maximum interface temperatures, heat pipe geometries, etc.

All of the thermal performance analysis results presented thus far in this report have been based on constant thermal output from the heat pipes. Such an assumption is suitable for much of the required analyses, since the data obtained is representative of the "average" system design and operation one would like to achieve. However, it is important, for the purpose of individual heat pipe design in particular, to have a model which describes transient system operation. Transient operation of the heat pipe waste heat removal system depends on header conditions, heat pipe characteristics, and properties of the rock heat sink. Knowledge of transient system operation allows one to design the system for control of start-up and excursions from normal operation.

A model for transient heat pipe operation was developed in the previous section on heat pipe theory. This development was accomplished by combining finite difference analysis techniques with Equation (C-36). Equation (C-36) is the transient heat pipe energy balance which is based on the header fluid bulk temperature, the temperature at the heat pipe and rock interface, and the thermal properties of the heat pipe. This model couples the thermal characteristics of the three major components of the heat pipe waste heat removal system; the waste heat source fluid header, the heat pipe, and the rock heat sink.

The transient heat pipe model was used to generate the data plotted in Figures 17 and 18. Figure 17 shows the heat pipe and rock interface temperature as a function of time for two different types of heat pipes; 1) a carbon steel evaporator and condenser and, 2) a stainless steel evaporator and a rubber condenser. The waste heat source fluid is water at T= 210°F with a film heat transfer coefficient h_e of 176 BTU/hr-ft^2-F and the rock has a thermal conductivity K_R of 1.6 BTU/hr-ft-F and an initial temperature T_R of 100°F. The heat pipe working fluid is water. Figure 18 is a plot of the condenser heat output as a function of time for the same system operating parameters. Both of these figures are semi-logarithmic plots with time being on a $\log_{10}$ scale.

It may be noted in Figure 17 that the interface temperature for both types of heat pipes, rises fairly rapidly during the first ten hours of operation, slows to a moderate pace during the time period 10 to 100 hours, and changes very little over the last 9,900 hours. The interface temperature remains well below the anticipated heat pipe operating temperatures at all times for the carbon steel heat pipe, but approaches these temperatures for the stainless steel/rubber heat pipe.

Referring to Figure 18, it is seen that condenser heat output, in the same manner as interface temperature, varies rapidly during the first few days of operation. In fact, the heat output reached a maximum in less than an hour, decreases rapidly until approximately 100 hours of operation, and then remains relatively constant thereafter. The heat output of the carbon steel heat pipe remains above 5 KW (17,065 BTU/hr) at all times up to 10,000 hours, but the heat output of the stainless steel/rubber heat pipe decreases below this value at approximately 100 hours.

The above described results shed some light on the value of the heat pipe/tunnel header system as a waste heat removal application for Deep Bases. However, they are primarily important as validation of the analytical procedures needed for thermal performance study of any Deep Base heat pipe waste heat removal application. These tools may be further refined, in cases where it is

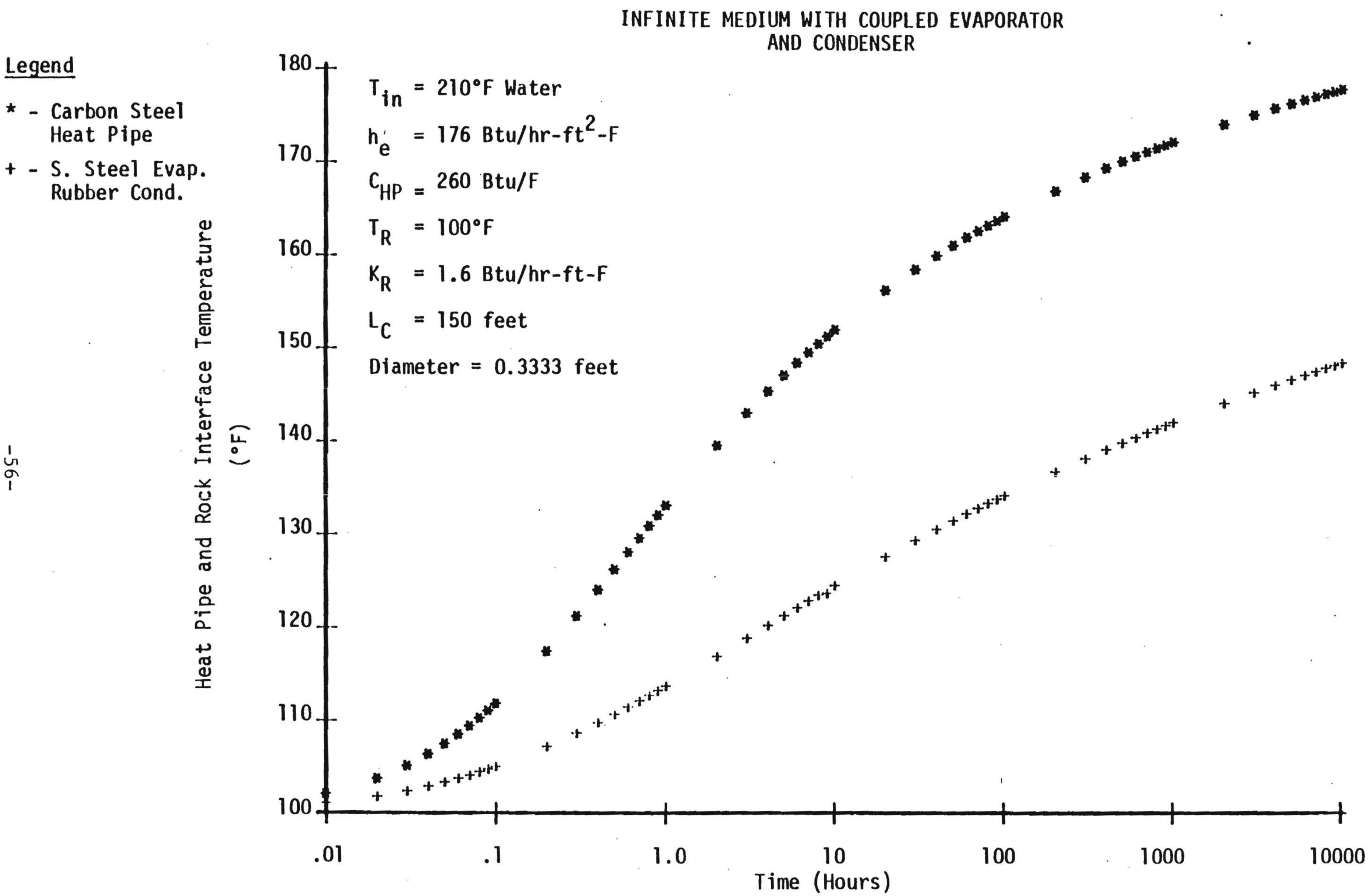


FIGURE 17

HEAT PIPE AND ROCK INTERFACE TEMPERATURE AS A FUNCTION OF TIME

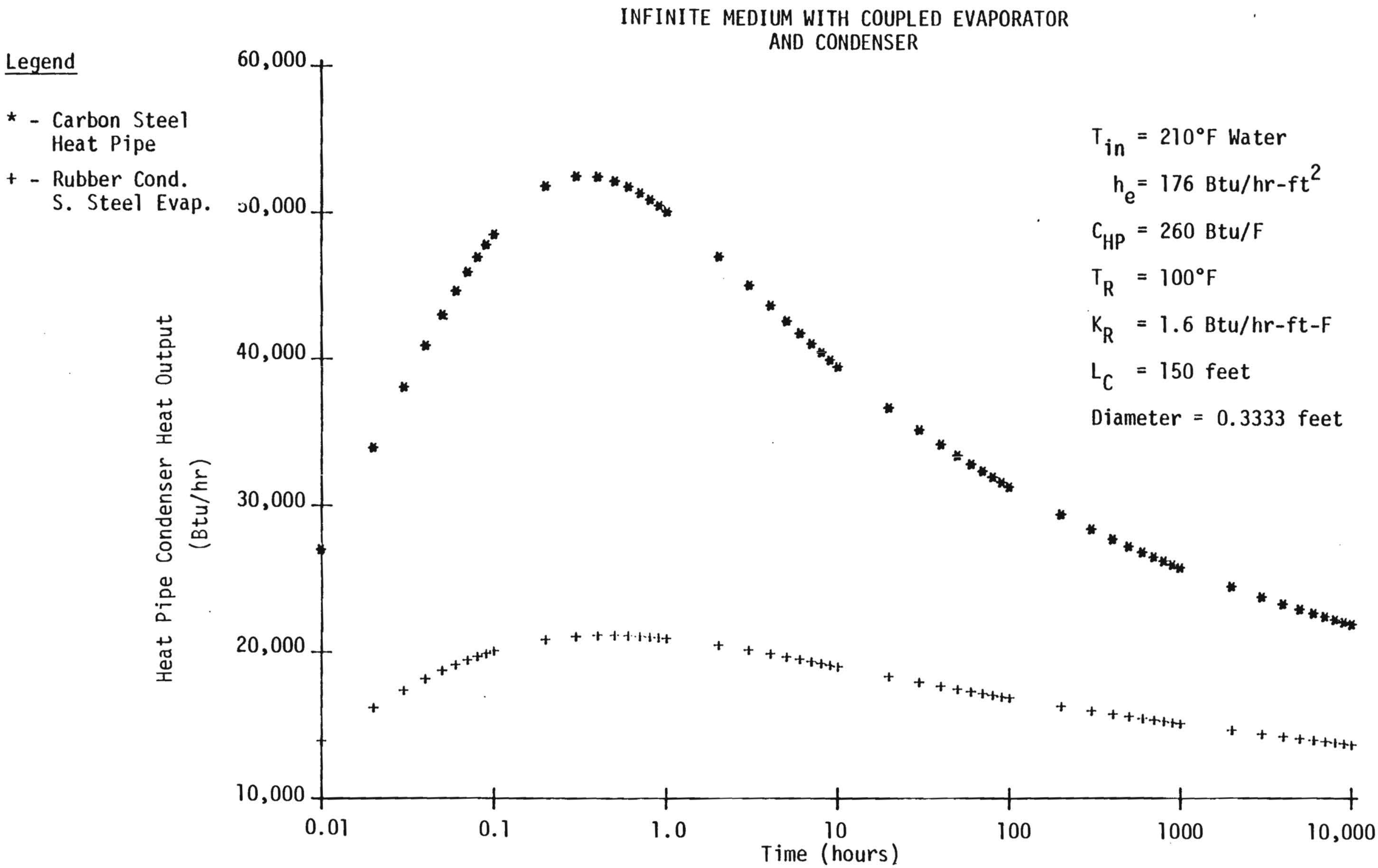


FIGURE 18

HEAT PIPE CONDENSER HEAT OUTPUT AS A FUNCTION OF TIME warranted, but in their present form they provide a firm basis for thermal analysis of the use of heat pipes to address specific waste heat removal problems that arise in the course of study of the Deep Base concept.

III. MECHANICAL PERFORMANCE ANALYSIS

Though this feasibility study has focused primarily on the thermal performance of heat pipe heat dissipation systems for Deep Bases, at least as important an issue is their mechanical performance. Major considerations related to mechanical performance are survivability, installation, performance, maintenance, and service life. All of these issues are examined in more detail in the following discussion.

A. Survivability

Survivability of the Deep Base heat pipe heat removal system is related to mechanical failures due to causes other than those which might be considered normal for a system of this type; e.g., mechanical wear and tear. Failures of this nature would arise primarily as a result of the system being subjected to forces due to natural geological processes or weapons effects. Careful siting of the Deep Base can help to minimize forces associated with geological effects so that they may not present a significant problem. The location selected may also provide some benefits related to attenuation of weapons effects; e.g., siting beneath a mesa. However, the entire Deep Base facility is expected to be subjected to significant forces resulting from surface detonations during the post-attack operations period and these forces represent the single greatest threat to heat pipe survivability.

The factors expected to have the most influence on heat pipe survivability are geometry (heat pipe dimensions and system configuration), materials of fabrication, and redundancy. Geometry and materials of fabrication will be intimately related while redundancy refers to system overdesign to account for unexpected operating excursions and non-repairable loss from service of individual heat pipes or, perhaps, an entire heat pipe module.

A detailed study of the importance of geometry to survivability cannot be performed at this time because of its dependence on the thermal performance analysis, which is only in its initial stages, coupled with a lack of knowledge about the expected environment (i.e., types and magnitudes of mechanical forces). However, the following example will serve to illustrate how this evaluation can be performed.

Figure 19 presents an isometric view of the undeformed model of a section of heat pipe. Hytrel 6346, a plastic, has been selected as the material of fabrication for the heat pipe condenser. This selection might seem surprising in view of the fact that plastics, in general, are poor heat conductors. The reason that such a material may be considered for use in this application is because the rock heat sink is such a poor heat conductor that it controls the overall heat transfer. This situation makes all other heat transfer resistances, even where poor conductors are used, less important by comparison. Therefore, a wide range of construction materials may be considered, including flexible materials such as plastics, which may be important for resistance to shock.

The model has a diameter of four inches and a wall thickness of 0.4 inches. The length considered is one and a half times the diameter or six inches. The model is divided into square elements 0.3 inch on a side to facilitate finite element analysis of the effect of forces using the GTSTRUDL system. GTSTRUDL is a computer aided structural engineering software system maintained by the School of Civil Engineering at Georgia Institute of Technology to assist engineers in the structural analysis and design process. This analysis tool is a fully integrated general purpose structural information processing system capable of supporting an engineer with accurate and complete technical data for design decision making. An overview of GTSTRUDL is given in Appendix B.

The event which the model experiences is a two-element horizontal separation of the surrounding rock and a one inch vertical displacement of the rock faces. An isometric view of the deformed model after occurrence of the event is shown in Figure 20 and a side view presented in Figure 21.

The event analyzed in this example has the characteristics of minor block motion along a fault plane. This particular event was selected for an example because it could easily represent one of a number of events which could possibly

MODEL

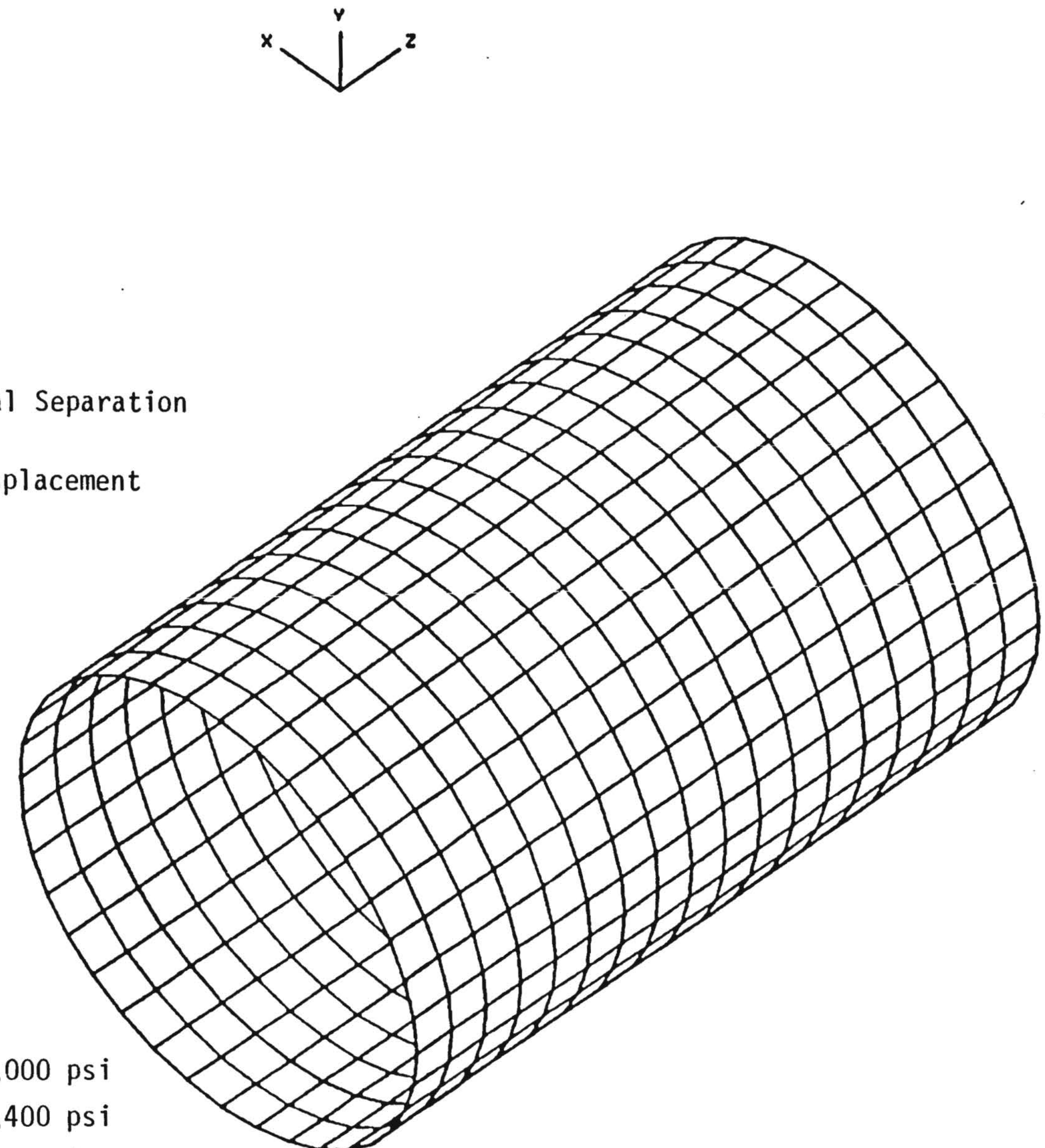
D = 4"

L = 1.5D

t_w = 0.4"

Two Element Horizontal Separation of rock

One Inch Vertical Displacement of Rock Faces



MATERIAL

Hytrel® 6346

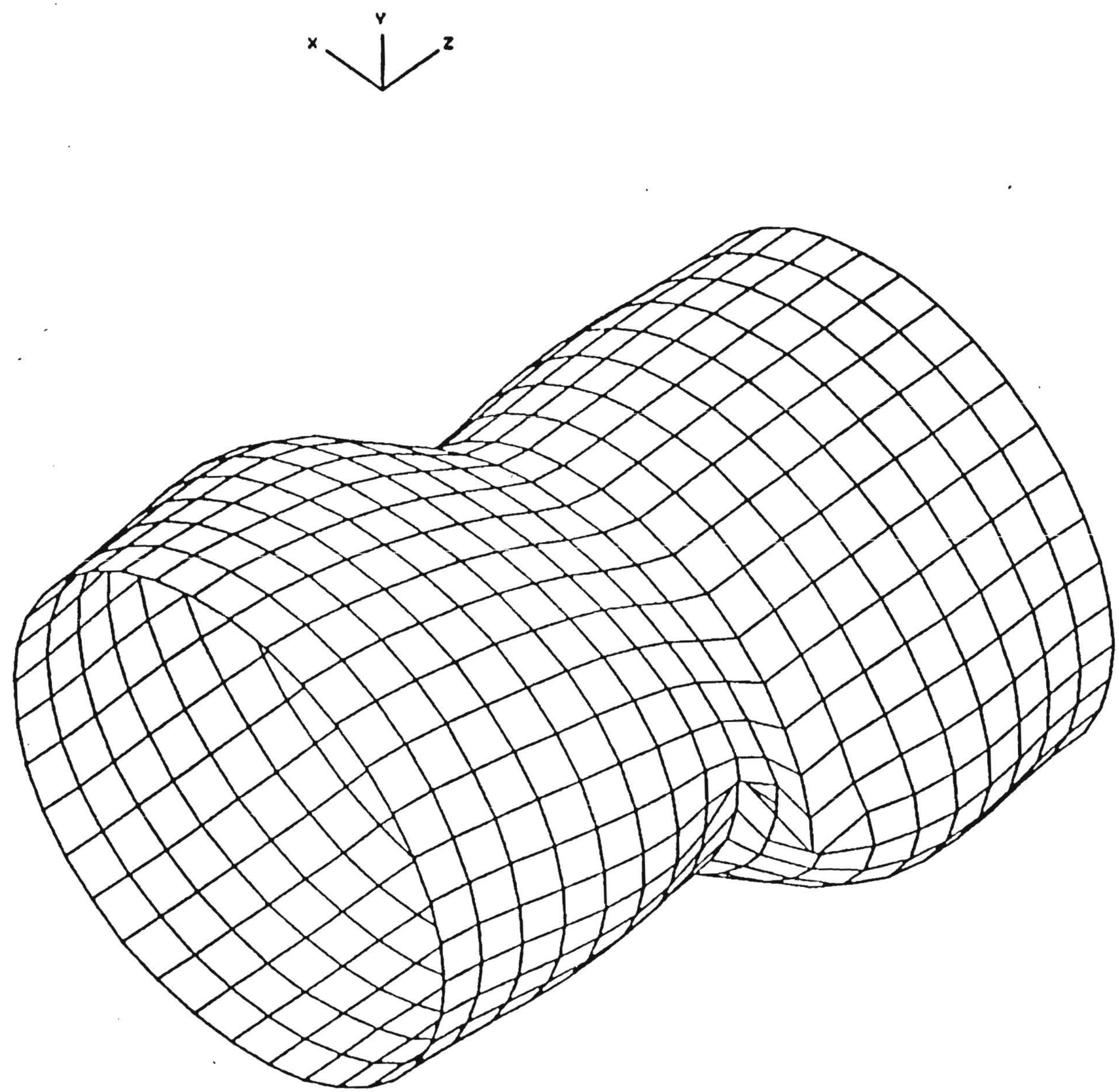
Elastic Modulus = 13,000 psi

Tensile Strength = 6,400 psi

Shear Strength = 6,950 psi

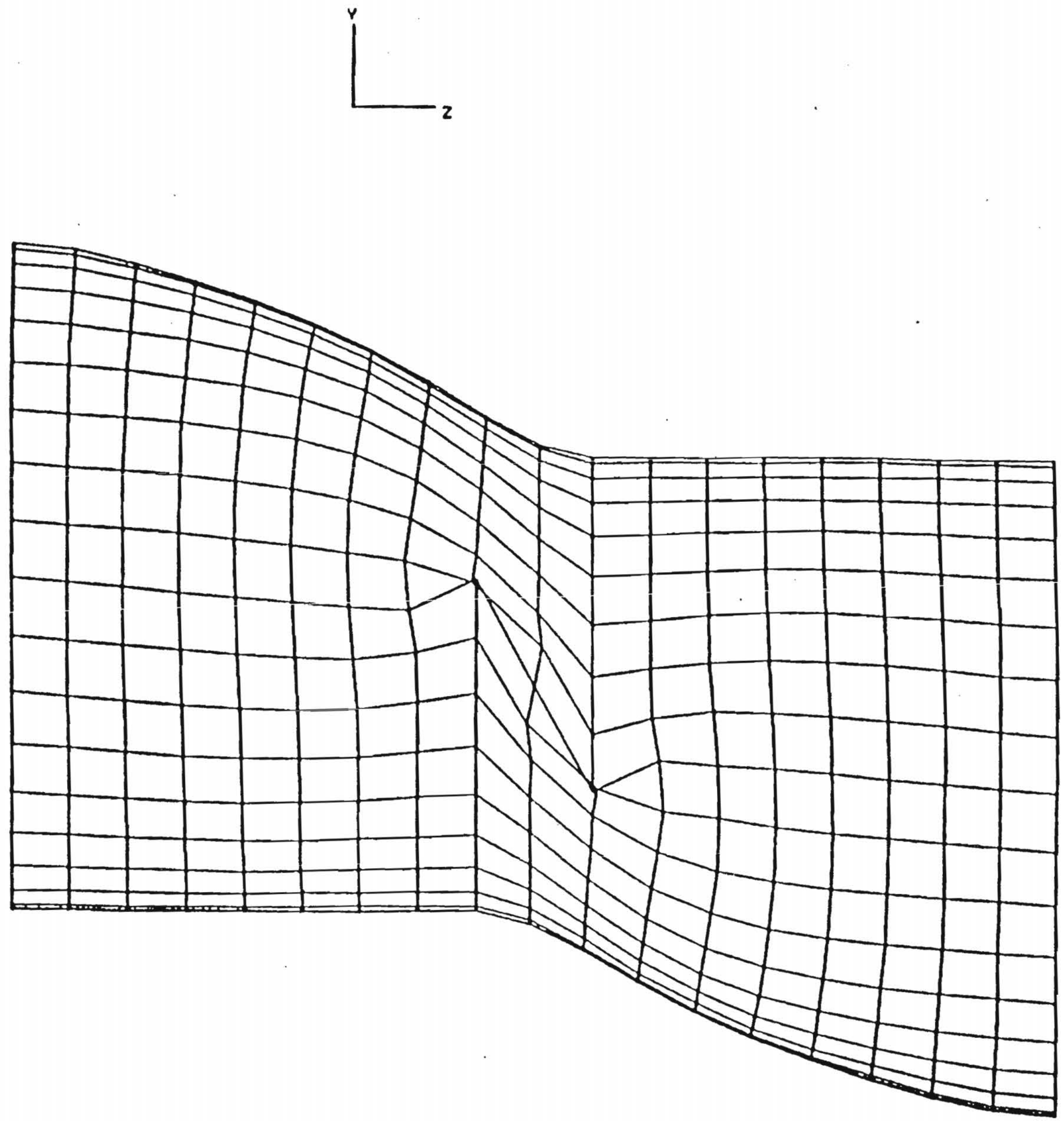
ISOMETRIC VIEW OF UNDEFORMED MODEL

FIGURE 19



ISOMETRIC VIEW OF DEFORMED MODEL

FIGURE 20



SIDE VIEW OF DEFORMED MODEL
FIGURE 21

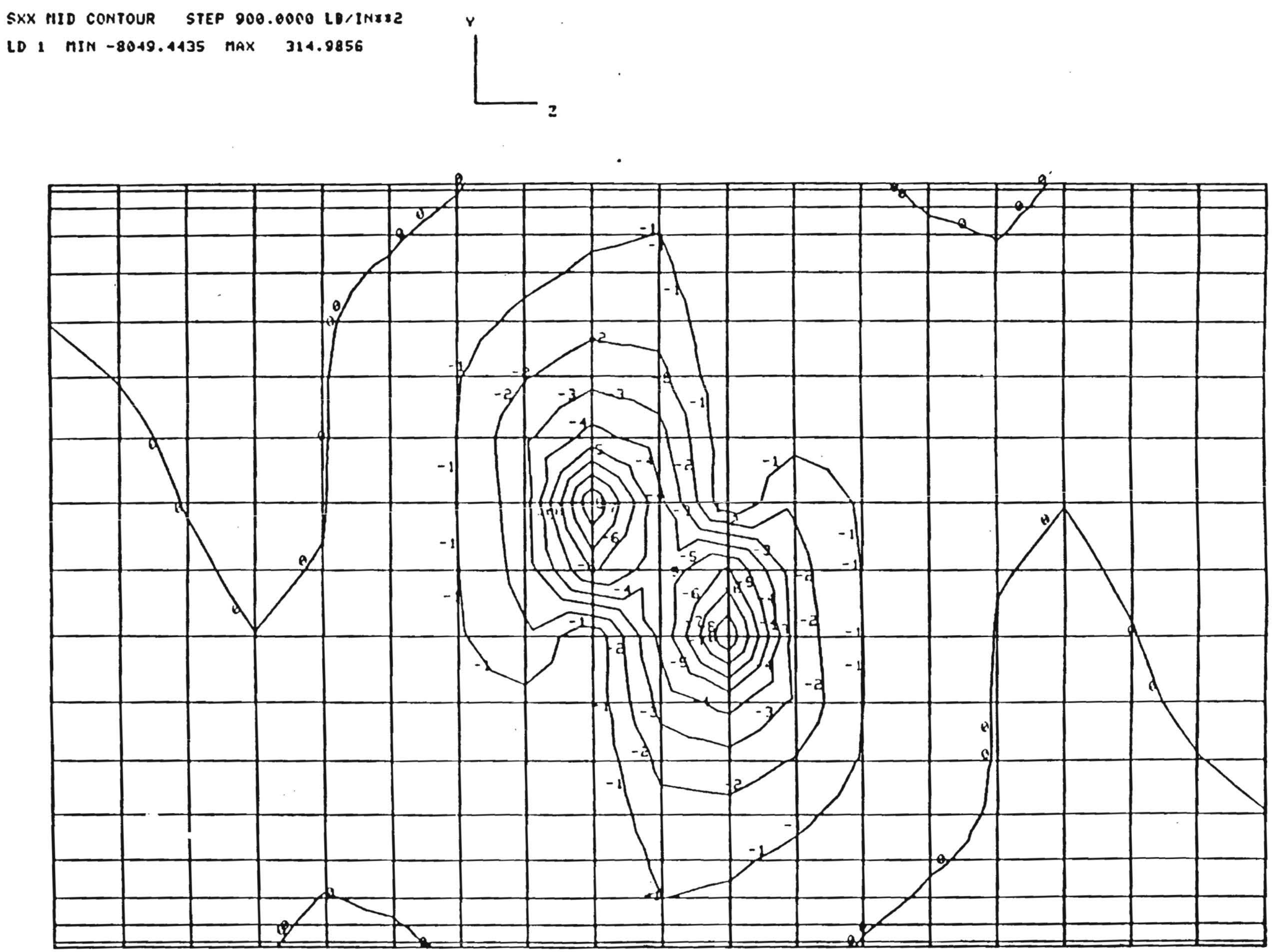
occur as a result of stresses induced in the surrounding rock by the tunnel boring operation used to build the Deep Base facility.

In order to assess quantitatively the effects on the heat pipe of the event, the GTSTRUDL program calculated mid-plane (center of tubing wall) normal and shear stresses in the tubing wall. The important stresses in this example are the normal stresses in the x- and y-directions and the shear stress in the x-direction on the y-plane (the reference coordinate system is that which is local to each element).

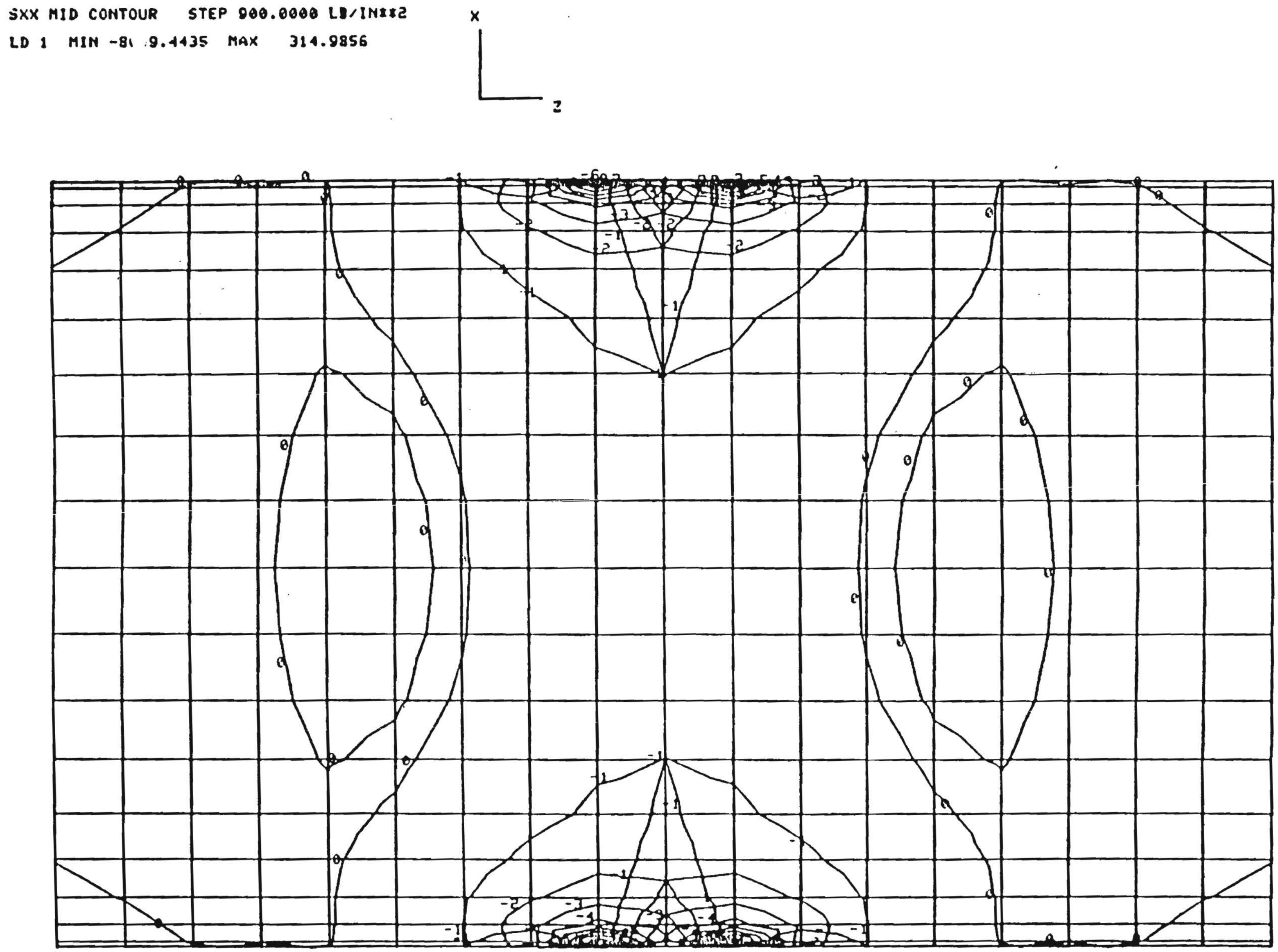
Figures 22 and 23 present side and top views, respectively, of the normal stresses in the x-direction while Figure 24 shows a side view of the normal stresses in the y-direction. Figure 25 presents a side view of the shear stresses in the x-direction on the y-plane.

In all of the normal stress plots, the integer numbers shown refer to a step value of 900 lb/in^2 (i.e., a value of one refers to a stress of 900 lb/in^2, a value of two equals 1800 lb/in^2, etc.). On the shear stress plot the step value is 700 lb/in^2. In all cases, maximum and minimum stress values are shown above the plot with a positive sign referring to tension and a negative sign to compression. A comparison of normal stresses calculated at the top-plane (outer edge of tubing wall) with normal x-stresses calculated at the mid-place yielded no significant differences in values, which indicates that mid-plane stresses are probably representative of the forces at any point in the tubing wall.

Even though the analysis of this particular case requires further refinement, which can be performed using GTSTRUDL, some interesting points may be noted. First, the maximum shear stress calculated (τ_{xy} = 6256 lb_f/in^2) is very near to the shear strength (6950 lb_f/in^2) of Hytrel 6346. Second, the maximum normal tensile stresses calculated (τ_{xx} = 315 lb_f/in^2, σ_{yy} = 820 lb_f/in^2) are well below the tensile strength (5400 lb_f/in^2) of Hytrel 6346. However, data are generally lacking for the compressive strength of many plastics, including Hytrel, so that it is difficult to determine if the maximum compressive stress calculated (σ_{xx} = 8049 lb_f/in^2) would cause failure of the heat pipe. It is also worth noting that a deformation of this type, depending

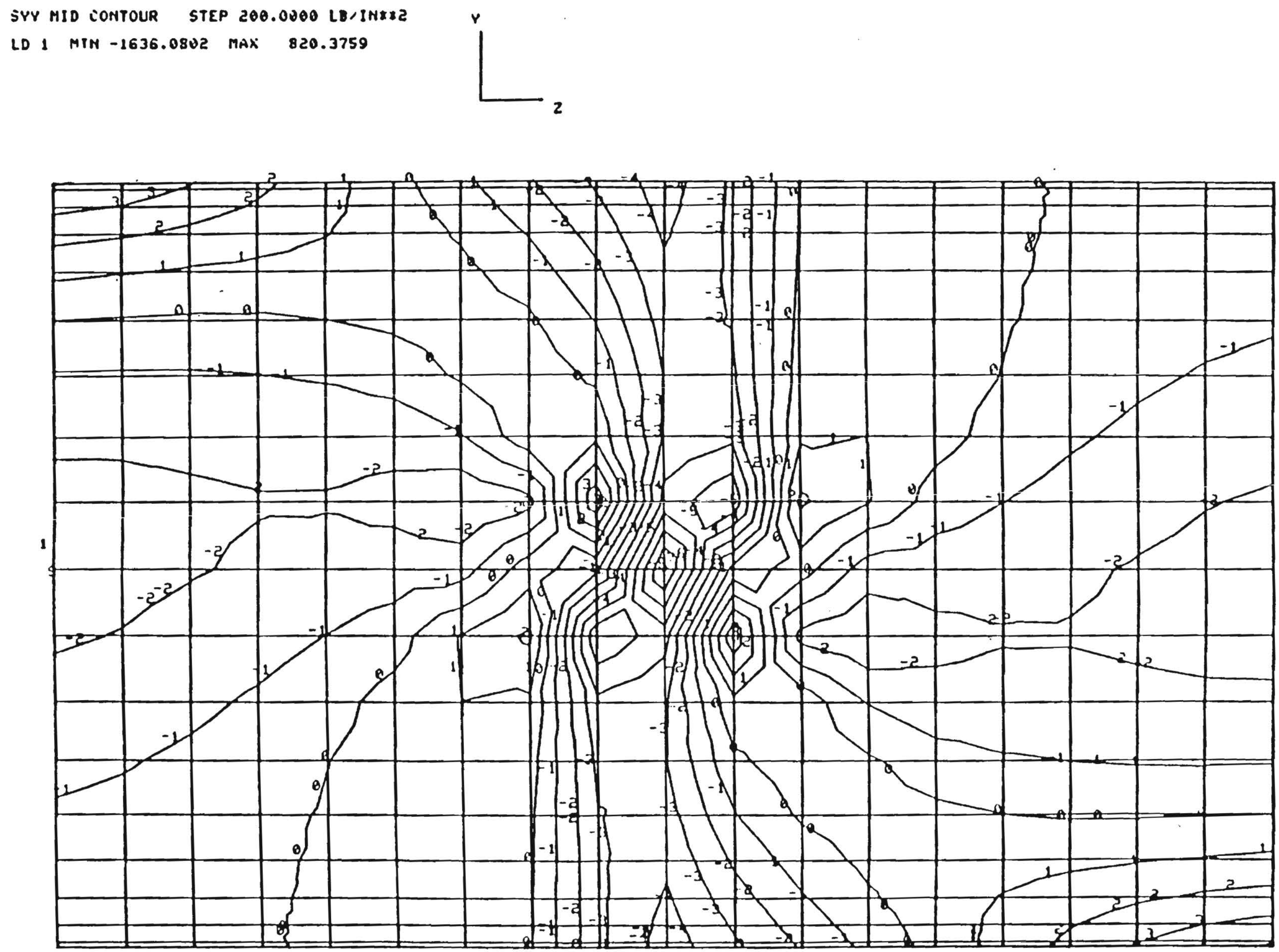


CONTOUR MAP OF MID-PLANE NORMAL STRESSES IN THE X-DIRECTION - SIDE VIEW
FIGURE 22

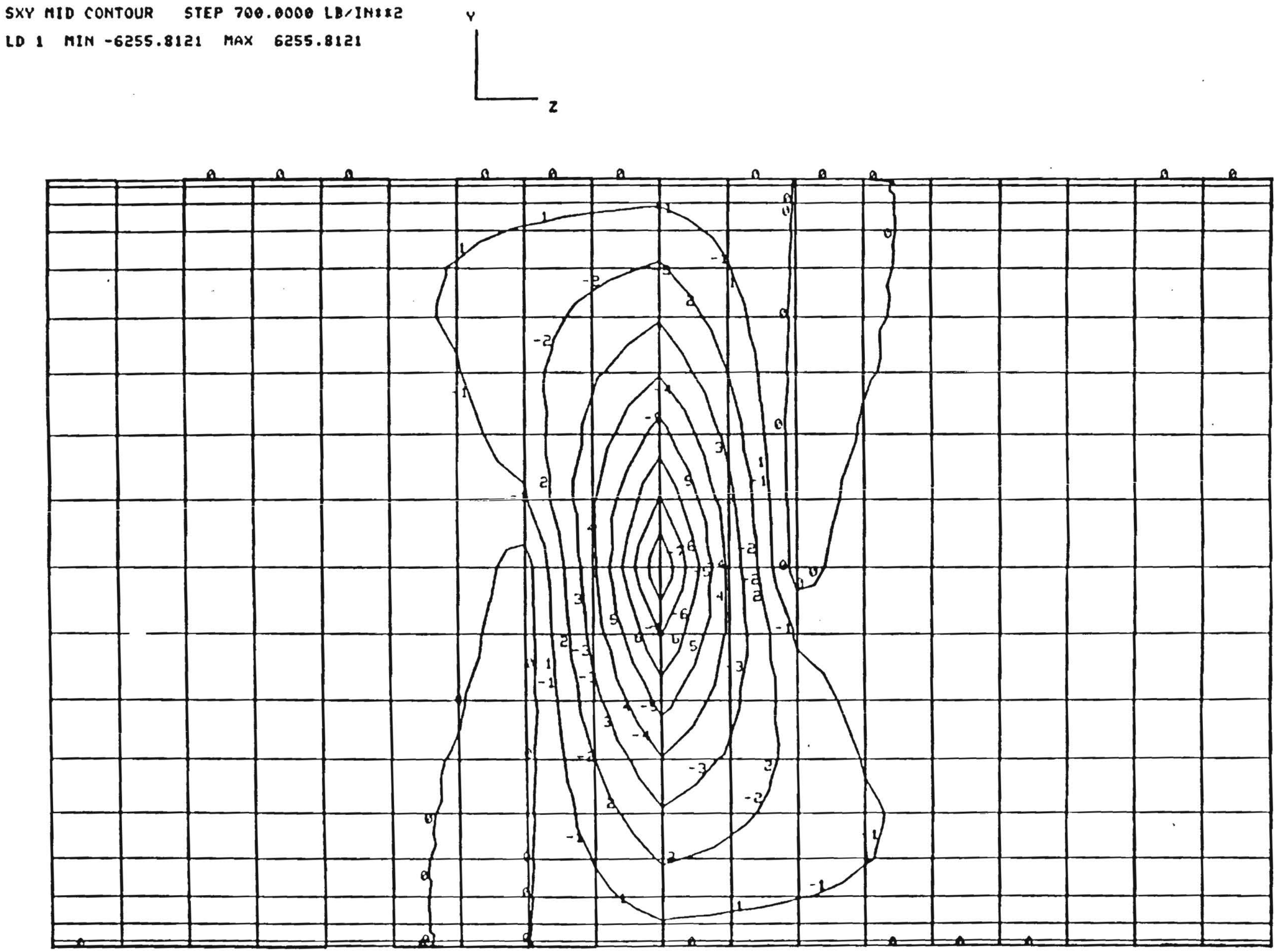


CONTOUR MAP OF MID-PLANE NORMAL STRESSES IN THE X-DIRECTION - TOP VIEW

FIGURE 23



CONTOUR MAP OF MID-PLANE NORMAL STRESSES IN Y-DIRECTION - SIDE VIEW
FIGURE 24



CONTOUR MAP OF MID-PLANE SHEAR STRESSES IN X-DIRECTION ON Y-PLANE - SIDE VIEW

FIGURE 25

on its location and orientation, could substantially affect the internal fluid dynamics of the heat pipe and, therefore, its thermal operation.

B. Installation and Maintenance

Because of the depth below ground level, limited size, and specialized requirements of the Deep Base tunnel facility, considerations related to the installation and maintenance of the heat pipe heat removal system will be very important. Further complexity will exist because of the requirements for high reliability and survivability. These factors taken in combination result in several significant technical problems which must be studied and overcome in order to insure successful utilization of heat pipes in Deep Bases. While not optimized and possibly not the exact configuration that might be used in a Deep Base, the conceptual design of a heat pipe/tunnel header waste heat removal module described earlier in this report is representative enough to be used as a basis for discussion of installation and maintenance considerations.

The schematic for the conceptual design of a heat pipe/header module is shown in Figure 2. Installation of the heat pipes in this design would require the drilling of holes 200-300 feet in length in a direction perpendicular to the axis of the tunnel. These holes would be 3-6 inches in diameter and be slightly inclined with respect to horizontal (10 degrees or less).

It is expected that the drilling of the heat pipe installation holes would be accomplished using existing horizontal drilling technology modified to operate within the size and orientation constraints of the Deep Base. A search of the literature identified a number of applications of horizontal drilling with much of this technology being tested and proven for more than ten years [16, 17, 18, 19, 20]. Table 4 is based on data taken from Reference [16] and shows actual and potential applications of horizontal drilling technology.

There are four candidate techniques for drilling long-range horizontal boreholes in rock. These include:

(1) Diamond wireline coring

TABLE 4

APPLICATION OF HORIZONTAL DRILLING TECHNOLOGY

Application	Typical Distance Penetrated, (ft)
1. Production Drilling	
Coal, degassing	1,000
Tar sands	Proposed
Oil shale (helical bore-hole path)	Proposed
Geothermal drilling	Proposed
2. Exploratory Drilling	
Reservoir investigation	3,300
Preliminary exploration of recovery regions	1,650
Drilling in coal seams	1,000
Preliminary exploration for tunnel construction	5,400
Underground explosions	3,300
3. Auxilliary Drilling in Civil-Engineering Projects	
Drainage	1,000
Grouting	1,000
Freezing holes	Proposed
Earth anchors	Proposed
4. Drilling for Utilities	
Cables	Proposed
Pipelines (gas, water)	Proposed

(2) Rotary drilling

(3) Down-hole motor drilling

(4) Down-hole percussive drilling

Figure 26, taken from Reference [17] illustrates the state-of-the-art of long horizontal drilling. In this case, state-of-the-art has been defined in terms of the capabilities of available production hardware and techniques which have been proven in horizontal drilling applications. Inherent in Figure 26 is some overlap of technologies and rounding off of numbers, but it still represents a realistic graphical summary.

Of particular interest is the fact that the hole diameters and lengths expected to be required for installation of heat pipes in Deep Bases are easily covered by existing horizontal drilling applications. Further, a list of vendors of horizontal drilling equipment found in Reference [18] was surveyed resulting in a compilation of information on equipment which might be suitable for use in Deep Bases. This vendor information indicates that horizontal drilling equipment is readily available which, if redesigned to satisfy the size and orientation constraints of the tunnel and system geometry, could be used to drill the heat pipe installation holes.

A similar problem to that of drilling horizontal, radial holes within the constraints of the Deep Base is the selection of the proper method for insertion of the heat pipes once the holes have been prepared. Since the Deep Base tunnel diameter is expected to be no more than 18 feet, the heat pipes, whose length will be on the order to 150 to 200 feet, will have to be inserted in sections or unrolled from a continuous coil.

Significant experience exists in the oil and gas production industry related to the insertion of continuous coiled steel tubing, on the order of two inches on diameter, into wells of more than 1000 feet in depth for the purpose of washing out debris. This technology could possibly be adapted to use for installation of heat pipes in Deep Bases. The size constraints of the tunnel would probably limit the use of this installation method to smaller diameter heat pipes than discussed in the thermal analysis section of this report if a

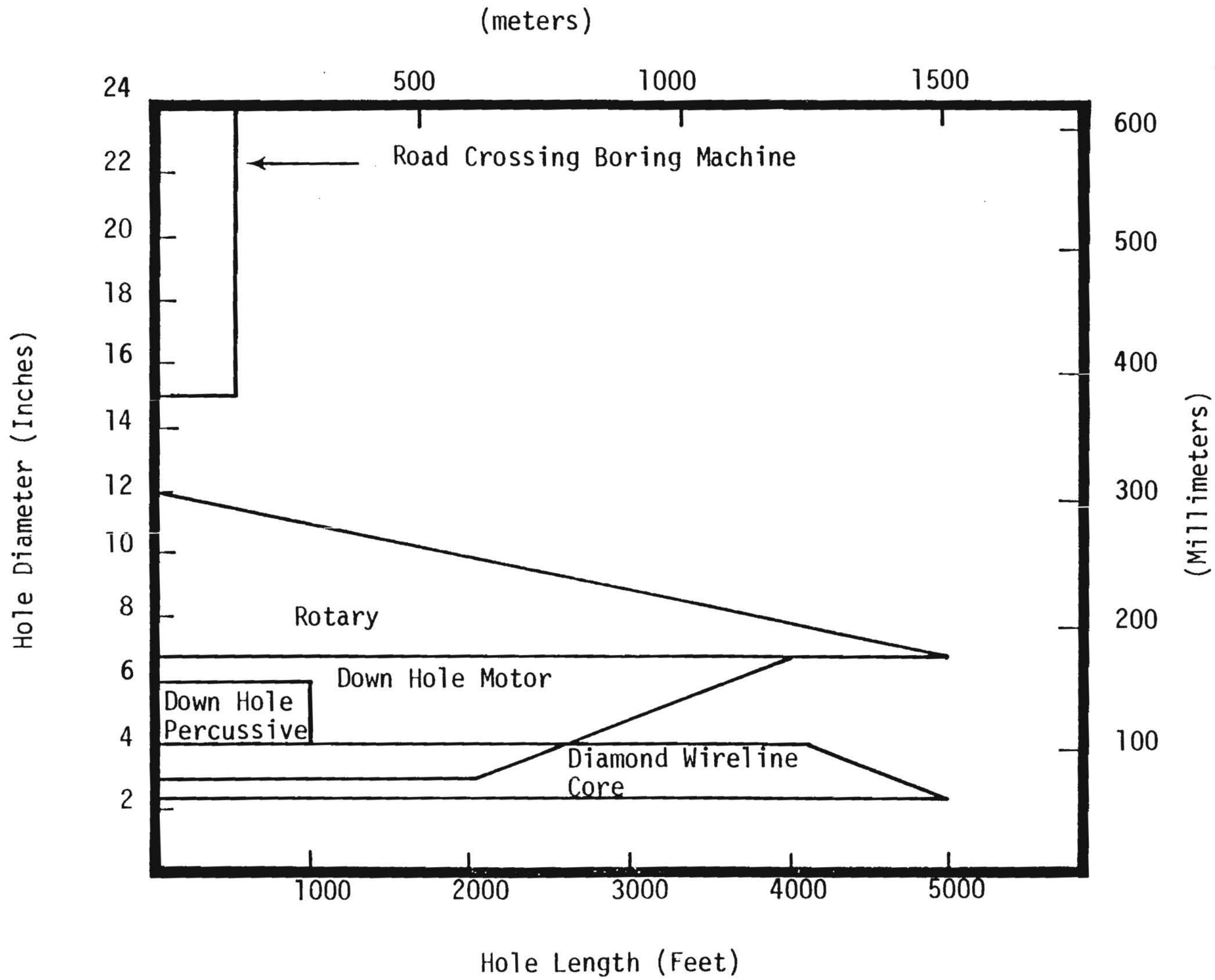


FIGURE 26

STATE-OF-THE-ART HORIZONTAL PENETRATION CAPABILITY metal is chosen as the material of fabrication. Use of a plastic or elastomeric material would probably allow one to go to larger diameter coiled tubing. However, this advantage would most likely be offset by added difficulty in overcoming "hang-up" and frictional resistance to insertion of the heat pipes into the drilled holes, a problem which would be exacerbated by the flexible nature of plastic or rubbery materials.

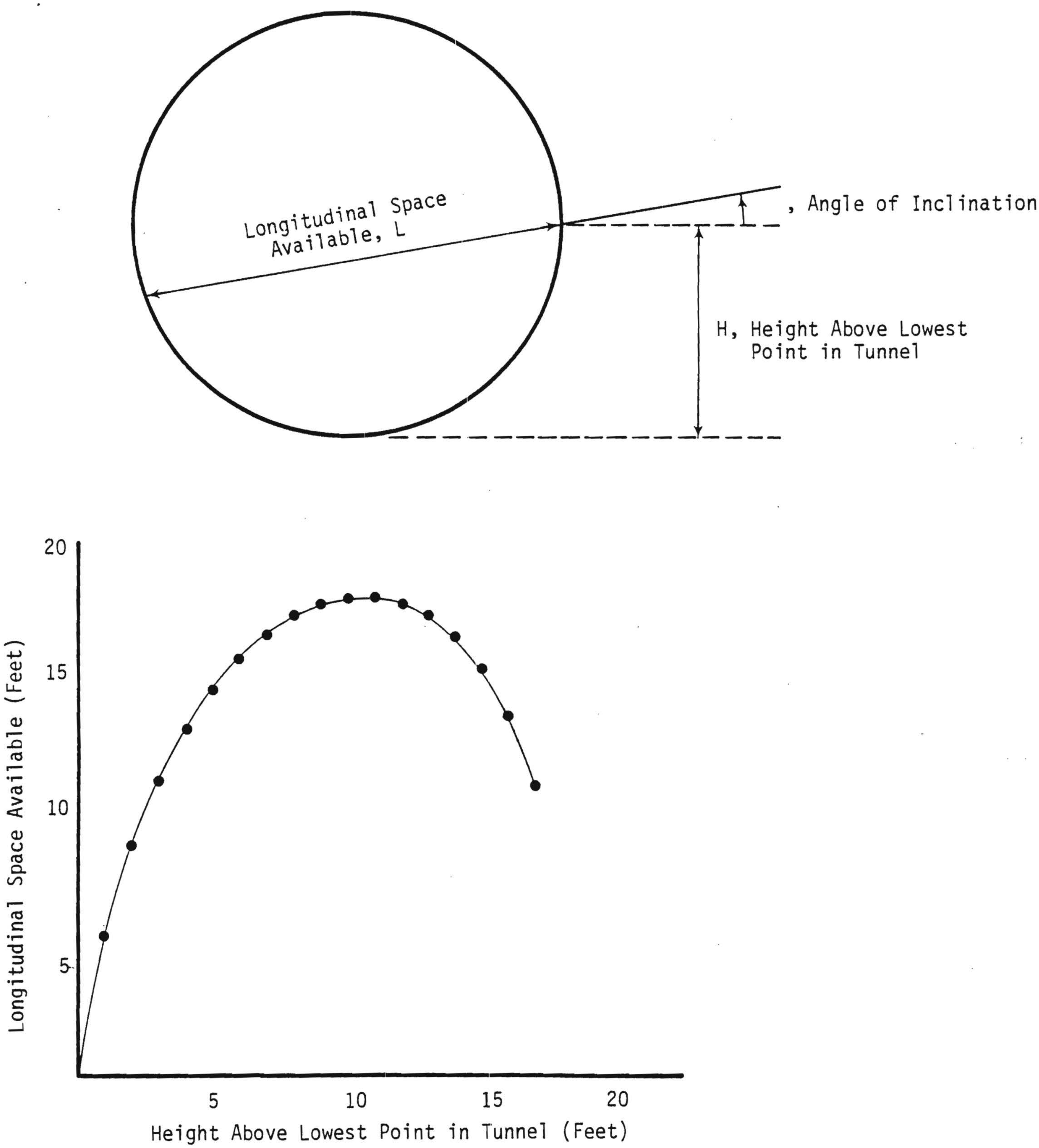
The most promising method for installation of heat pipes in Deep Bases is to insert them in sections. These sections could be joined to one another by threaded couplings or by welding, with either of these attachment methods capable of being performed automatically by machine.

If the heat pipes were inserted in sections, the maximum length for the sections of a particular heat pipe would be dictated by the position and orientation of that heat pipe in the Deep Base tunnel. For that matter, the heat pipe location would have already placed similar size constraints on the horizontal drilling equipment and the length of drill rod used. Figure 27 presents the longitudinal space available as a function of distance above the lowest point in an 18 foot diameter tunnel for a 10 degree elevation above horizontal and assuming the drill string or heat pipe is perpendicular to the tunnel axis. In this case, the space available to contain drill rod or heat pipe section has a value of 5.34 feet at a point which is one foot above the lowest point in the tunnel and increases to a maximum of eighteen feet (the tunnel diameter) at slightly more than than eleven feet above the lowest point and then decreases to 10.90 feet at seventeen feet above the lowest point (one foot below the highest point). The longitudinal space available for the drill string or heat pipe sections may be increased by decreasing the elevation above horizontal and by deflecting the direction to some angle less than 90 degrees (perpendicular) with respect to the tunnel axis.

Implicit in the installation concepts described above is the notion that the heat pipes will be fabricated in place. The entir- installation procedure would entail a number of distinct steps. First, the installation hole would be drilled to the required depth and orientation at a specified tunnel location. Second, heat pipe condenser sections with internal capillary structure would be

FIGURE 27

LONGITUDINAL SPACE AVAILABLE AS A FUNCTION OF HEIGHT ABOVE LOWEST POINT IN TUNNEL



simultaneously joined and inserted into the installation hole. Following insertion, the condenser would be cleaned thoroughly and plumbed to the evaporator which is expected to be fabricated ouside of the Deep Base and then shipped in. The entire heat pipe would then be leak tested, filled with the prescribed amount of working fluid, and sealed.

Even though the thermal resistance of the air gap between the outer surface of the heat pipe condenser and the hole is expected to be small compared to the thermal resistance of the rock, it could still adversely affect overall performance of the heat removal system. If required, this problem could be overcome by displacing the air with a material which is a good heat conductor; i.e., a "thermal grout." This "thermal grout" would assure good heat transfer from the heat pipe to the rock.

A material which is a promising candidate for use as the "thermal grout" is silicone heat sink compound of the type used in electronics applications; e.g., thermal coupling of an electrical/electronic device to the heat sink or chassis [21]. This material is a silicone thickened with metal oxide filler and has a greaselike physical form. Its special properties include high thermal conductivity, low bleed, and stability at high temperatures.

The "grout" could be put in place by pumping the volume required to fill the air gap into the heat pipe installation hole before the condenser is inserted. Then, as the condenser is inserted, the "grout" would be forced out thereby displacing the air and providing a positive thermal seal between the heat pipe condenser and the rock. Since the silicone heat sink material is a fluid it would also provide some cushioning effect against shock and rock movement which could enhance the survivability of the heat pipes.

Once all of the installation steps were completed, operation of an individual heat pipe could be tested by applying a known heat load to the evaporator and observing the ability of the heat pipe to efficiently remove it. Following the completion of an entire module of heat pipes, its performance could be checked in the same manner as a single heat pipe. Such tests, if performed routinely, could assure reliable operation of heat pipe modules allowed to stand idle and then suddenly called into service; e.g., upon initiation of post-attack "button-up."

Once the heat pipes were put into operation, their performance could be monitored in much the same way that system checkout was accomplished. By knowing the heat load being applied to each module at all times, its ability to dissipate heat could be determined giving an indication of both slow deterioration in performance or catastrophic loss in capability, as the case might be.

Once a problem module was identified, individual heat pipes could be checked to determine which ones were operating poorly and causing performance loss. In many cases, problem heat pipes might be repaired simply by repeating some of the same steps which were used for initial installation. For example, a heat pipe could be disconnected at the evaporator and the fabrication steps of cleaning, evacuation, leak testing, and filling with working fluid repeated. In more extreme cases, the heat pipe condenser could be pulled from its installation hole and repaired or a new condenser inserted. For cases where heat pipe is non-repairable, it might even be possible, depending on the location of the heat pipe in the tunnel facility, to drill an entirely new installation hole and refabricate a replacement heat pipe.

An important characteristic inherent in heat pipe waste heat removal system being studied here is that the deterioration in performance or loss of a few of the heat pipes from service will not be detrimental to overall system performance. This characteristic is a result of the dispersed nature of this heat removal system and the fact that the design of the individual heat pipes can be made such that the operating heat pipes will pick up the increased heat load caused by the heat pipes which are out-of-service or operating with reduced efficiency.

Another valuable feature of this heat removal concept arises from the fact that it utilizes the rock surrounding the Deep Base as the heat sink rather than depending on one which is artificially created. Artificially created heat sinks will have finite thermal capacities, which will be exhausted within a planned period, while the sheer magnitude of the available rocks allows it to subsist as a heat sink for extended durations (essentially indefinitely with respect to the expected life of the Deep Base). There also exists the possibility of creating new heat sink capabilities by installing additional heat pipes in virgin rock for the purpose of heat sink replacement or increasing Deep Base size or longevity.

C. Service Life

The Deep Base facility is expected to operate in the pre-attack mode for a substantial period of time. In fact, since the function of this facility is to be a deterrent, the duration of this operational mode could be for its entire life which could easily be in excess of 20 years. Therefore, it is incumbent upon all of the Deep Base systems, including those for waste heat removal, to have a service life of at least this duration.

The heat pipes in the Deep Base environment will be subject primarily to three non-attack related influences on their service life: thermal degradation, chemical degradation, and permeability to non-condensable gases or working fluid vapor. Fortunately, all three of these factors can be designed for fairly easily in Deep Base waste heat removal applications.

The expected operating temperatures for the heat pipes are moderate, since they will be on the order of 150-200°F. At these temperatures, thermal degradation of most candidate heat pipe materials would be mild, even over the extended timeframe of 20 years or more.

Since it is anticipated that the rock environment will be dry or contain, at most, only water, chemical degradation would arise solely from attack by the working fluid or heat source fluid in the header. The working fluids which will be suitable for most of the Deep Base waste heat removal applications are water and methanol or, in special cases, freon or a freon mixture. The most probable header fluid would be water or again, in a special circumstance, freon. All of these fluids are relatively innocuous and not expected to cause significant chemical degradation, even over extended time periods, for most candidate heat pipe materials.

The final non-attack factor affecting heat pipe service life is permeability to gases and vapors. The presence of even minute amounts of non-condensable gases can severely reduce the film heat transfer coefficient in condensing processes. Therefore, leakage of non-condensable gases into the heat pipe could adversely affect its performance.

Fortunately, however, the internal fluid dynamics of the heat pipe serves to minimize the effects of non-condensable gases. Even at low vapor velocities for the working fluid these gases will be swept to the end of the condenser and held there in a compressed state during operation;. Therefore, one only has to overdesign the length of the condenser section just enough to account for loss of heat transfer area due to storage of the total estimated permeation of non-condensable gas over the life of the heat pipe. Even in the event that a large build-up of non-condensables causes a reduction in heat pipe performance below that considered acceptable, remedial action may be taken by disconnecting the heat pipe at the evaporator and repeating the fabrication steps of cleaning, evacuating, leak testing, and charging with working fluid.

Permeability of condensable vapors is important because it could result in significant loss of working fluid. Because of this factor, selection of the heat pipe condenser material must be done carefully. Metals are relatively impermeable to both gases and vapors and, therefore, if selected, should perform well. Some plastics and elastomers, however, even though relatively impermeable to non-condensable gases, will show high permeability to water vapor and some organic vapors. And to make matters worse, this permeability increases significantly with temperature. Therefore, if a plastic or elastomer is being considered as the heat pipe condenser material, its permeability characteristics should be studied very carefully before a decision is made to use it.

It should be noted that it is expected that the driving force for permeation, which is the pressure difference across the heat pipe wall, will be low during both operational and idle periods for the heat pipes. While the heat pipes stand idle, a vacuum will exist in their interior. Therefore the maximum theoretical pressure difference across the heat pipe wall will be one atmosphere resulting in a relatively low tendency for non-condensable gases to permeate into the heat pipe. The heat pipes will operate at internal pressures which are on the order of one atmosphere resulting in low driving force for permeation of working fluid vapor out of the heat pipe.

Perhaps the most important result of the service life characteristics described above is that, barring catastrophic failure due to attack related phenomena, a heat pipe waste heat removal system will have graceful degradation in performance over the life of the Deep Base. The non-attack related factors affecting heat pipe service life are expected to result minimal deterioration in performance and relatively few heat pipes lost totally from service. In addition, the operating heat pipes, by their very nature, will have the capability to compensate for loss from service or deteriorated performance of individual heat pipes so that effect on overall system performance will be negligible.

D. Heat Pipe Materials

As indicated in the Phase I final report for this concept feasibility study, mechanical performance characteristics, as well as thermal performance characteristics, should be used as the basis for selecting the material for fabricating the heat pipe condenser. The controlling thermal factor for any Deep Base heat removal system, including heat pipes, which dumps heat to the surrounding rock, is the heat transfer in the rock itself. Because this heat transfer is relatively poor, a material of construction may be selected for the heat pipe condenser section which has relatively poor thermal characteristics without adversely affecting overall thermal performance of the heat removal system. With thermal characteristics being somewhat less important, the major issues related to heat pipe materials selection could well be survivability, ease of installation and maintenance, life, and cost. Therefore, materials selection should focus on maximizing system survivability, ease of installation and maintenance, and life, while minimizing its cost.

The reduced importance of thermal characteristics for the heat pipe condenser material increases the number of candidate materials which may be considered. Potential heat pipe materials cover a broad range from very flexible elastomers through flexible but rigid plastics to very rigid, but not brittle, metals. In addition, it is possible that two or more of these candidate materials may be combined to form a composite which has the desired characteristics. A preliminary listing of the properties of representative candidate materials was presented in tabular form in the Phase 1 final report. This information is shown again in Table 5. Consideration of this wide variety of materials provides a broad range of mechanical performance characteristics which might be possible for the heat pipe waste heat removal systems.

A great deal more thermal and mechanical performance study will be required before the list of candidate materials can be narrowed down, much less an actual fabrication material selected. However, two important points may be made at this time. First, heat pipes have traditionally been constructed of metals because of their higher thermal conductivity and a metal will be the material of choice if mechanical performance and cost considerations can be satisfied. In fact, the evaporator will certainly be fabricated from high thermal conductivity metal since the thermal resistance between the waste heat source stream and the heat pipe working fluid should be kept as low as possible and mechanical performance requirements of this part of the heat pipe can be easily met with a metal.

The second point of interest is that, as indicated earlier in the installation and maintenance discussion, the capillary structure will be fabricated as an integral part of the tubing used to construct the condenser. This fabrication may be accomplished either through machining, forming, or etching the inside surface of the tubing or by attaching a cloth, plastic or metal mesh wick to the inside surface of the tubing, with this wick possibly performing a structural as well as thermal role. Fabrication of the capillary structure in this manner will not adversely affect heat pipe performance because pumping of the working fluid from the condenser back to the evaporator is accomplished by gravity and not by capillary action. The capillary structure, in this case, merely provides for better distribution of the condensing working fluid around the inside surface of the heat pipe condenser.

TABLE 5

RANGE OF CANDIDATE HEAT PIPE CONDENSER MATERIALS

	Hypalon[1] (chlorosulfonated Polyethylene)	Hytrel[1] (Polyester elastomer)	Nordel[1] (ethylene-propylene-diene polymer)	Vamac[1] (ethylene/acrylic elastomer)	Polyethylene	Polypropylene	Teflon[1] (polytetrafluoroethylene)	Polybutylene	Ryton[2] (polyphenylene sulfide)	Kynar[3] (polyvinylidene fluoride)	Copper 12200 DHP (deoxidized high Phosph. Alloy)	Aluminum 3003 - 0 (Soft tempered Al alloy)	Steel AISI 1010 HR
Tensile Strength, psi	2500	6400	3000	2500	2000-4000	5000	2500-3500	3000-4500	9500	6500	32,000	16,000	47,000
Yield Strength, psi					-	-	-	-	-	-	9,000	6,000	26,000
Specific Gravity	1.12-1.28	1.20	0.83	1.08-1.12	0.94-0.96	0.89-0.91	21-2.3	.925	1.3	1.8	8.8	2.75	7.86
Elongation, %	100-300	25% Yield 600% break	100-300	100-300	25-400	500-700	250-350	200	-	25-500	40	30	28
Temperature Limit, °F (upper limit cont. service)	275	230	293	329	250	275-320	500	200	450	280° (@ 220 psi)	400	400	-
Thermal Conductivity	.08 (4)	.08 (4)	.08 (4)	.08 (4)	0.19 (4)	0.08 (4)	.14 (4)	.125 (4)	.165 (5)	.065 (4)	117 (4)	224 (4)	25 (5)
Permeability to Gases	Low- V.Low	Fair	Fair	V. Low	Low	Low	Low	Low	Low	Low	None	None	None
Adhesion to Metals	Exc.	Good	Good-Exc.	Good	Poor	Poor	Good	Unknown	-	Good	N/A	N/A	N/A
Adhesion to Fabrics	Good	Outst.	Good	Excellent	Poor	Poor	Unknown	Good	-	-	N/A	N/A	N/A
Tear Resistance	Fair	Outst.	Good	Good	Good	Good	Good	Good	-	Good	Excellent	Excellent	Excellent
Abrasion Resistance	Exc.	Very Outst.	Exc.	Good	Good	Good	Good	Very Good	-	Very Good	Excellent	Excellent	Excellent
Compression Set	Fair	Fair	Good	Good	-	-	-	-	-	-	N/A	N/A	N/A
Cost per Pound	$1.50	$2.75	$1.20	$2.20	$0.75	$0.75	$5.75	$1.16	$4.05	$6.05	$1.40	$1.05	$0.50
Additional Data	Thermoset	Thermoplastic	Thermoset	Thermoset		Methanol resistance 140°F-Satisfactory 212°F-not suitable							

(4) Units are BTU/hr ft²°F/ft
(5) Units are BTU/hr ft°F

[1] Trade name DuPont Company
[2] Trade name Phillips Petroleum
[3] Trade Name Pennwalt Corp.

IV. PRELIMINARY SYSTEM COST CONSIDERATIONS

Detailed heat pipe waste heat removal system cost estimates are impossible to make at this point since both thermal and mechanical performance analyses are only in their initial stages. In fact, accurate assessment of system costs cannot be performed until the current study moves beyond the concept feasibility phase and specific Deep Base applications for heat pipes are chosen for evaluation. However, it is possible to consider some preliminary cost implications based on knowledge gained thus far about potential systems.

First, installation of heat pipe waste heat removal systems are expected to require minimal lengths of dedicated tunnel. This characteristic is in contrast to some of the alternative Deep Base waste heat removal concepts being considered which require the boring of, potentially, miles of additional tunnel to house a heat sink (e.g., ice/water tunnels) or provide heat transfer surface for condensing the waste heat source fluid (e.g., steam tunnels). The heat pipe/tunnel header concept considered in this study is planned for installation in portions of the tunnel facility which also have other uses. The functional role of the tunnel and possible coexistence with other Deep Base equipment could influence the configuration of the heat pipe system (e.g., placement of the header and heat pipes to accommodate other Deep Base support systems requirements) but are not expected to severely constrain its application.

The drilling of small diameter holes (relative to the anticipated tunnel diameter of 18 feet) will be required to install the heat pipes. As discussed in the previous section on mechanical performance, the drilling of these holes would be based on existing horizontal drilling technology and would be accomplished using commercially available equipment adapted to meet the specialized constraints of the Deep Base. Costs for this type of drilling are not readily available, but typical total costs for rock quarry drilling are on the order of 1.2 $/ft for 3.5 inch diameter, 42 feet long holes [22]. Horizontal drilling costs in the Deep Base are expected to be greater than this, but not a full order of magnitude greater.

The materials being considered for use in the heat pipe waste heat removal systems are, in general, commercially available. It is anticipated that the heat pipe condenser will be fabricated from a common metal such as carbon steel or, if the situation warrants, a readily available plastic or rubber material. Pipe made of such materials and in the diameters being considered are an of-the-shelf item and can be joined in a variety of ways; e.g., threaded coupling, welding, cementing, etc. The fabrication of an integral capillary structure will, of course, add to the cost of this pipe, but this increase is not anticipated to be large.

The evaporator is expected to be fabricated from a relatively high conductivity, but inexpensive metal, as an integral part of the header system using conventional heat exchanger fabrication techniques. The adiabatic section can be made of an inexpensive, flexible plastic or rubber or a metal bellows.

A cost which will have to be looked at carefully is that of the "thermal grout" since significant amounts of this material could be required. The silicone heat sink compound suggested as a possibility is quite expensive ($8.41/lb) for grades suitable for electronic applications [23]. However, Deep Base heat pipe applications will not require that this material have the quality that is required for use in electronics resulting in lower cost. In addition, if large quantities of this material are used, the improved economics of large-scale production should work to hold its price down. Other types of less expensive "thermal grout" materials may also be considered. And finally, it is also feasible that thermal performance analysis will show that the influence of the air gap between heat pipe condenser and the rock surface is negligible compared to other thermal resistances so that a "thermal grout" is not required at all.

Because experience with in-place fabrication of large heat pipes is non-existent, such costs cannot be estimated for Deep Base heat pipe applications at this time. These cost estimates will have to await the results of experimental and prototype studies planned for later, follow-on activity to the current feasibility study. Components of these studies will specifically address the development of in-place fabrication techniques and should result in all of the information needed to make reasonable cost estimates.

Finally, study thus far has indicated that a relatively small number of heat pipes will be required to dissipate anticipated Deep Base waste heat loads. Preliminary calculations show that as few as 200 heat pipes might be required to remove one megawatt of waste heat. The number of heat pipes required will, of course, greatly influence the total cost of the waste heat removal system.

V. CONCLUSIONS AND RECOMMENDATIONS

A feasibility study involving concepts in which heat pipes are used to transfer waste heat from a Deep Base to the surrounding rock has been performed over the past year for the Ballistic Missile Office of the United States Air Force. This effort was undertaken by the Technology Applications Laboratory and the School of Mechanical Engineering at the Georgia Institute of Technology. The study has shown that the use of heat pipes to remove waste heat from a Deep Base and dissipate it into the surrounding rock is a feasible and promising concept and, in fact, may be the best alternative to consider in some cases. In other instances, heat pipes might be effectively used to augment heat removal by alternate technologies, thereby improving the overall efficiency and reliability of Deep Base waste heat removal systems.

Examples of Deep Base waste heat removal applications that have been considered in this study and found to be attractive enough to merit further study include:

- Power Plant Waste Heat Removal
- Equipment Cooling
- Emergency Cooling

The conceptual design approach taken in this study emphasized operational flexibility for the heat pipe waste heat removal system as a whole. This feature is facilitated by designing modules which will handle some discrete portion of a particular waste heat load. Modules would be equipped with single or multiple heat pipes with varied geometries and working fluids to suit each specific application. This approach not only maximizes system operational flexibility, but also results in greater reliability and survivability.

Phase 1 of this concept feasibility study, reported on earlier, was directed at determining if there were significant technological or economic barriers which would prohibit the use of heat pipes for Deep Base waste heat removal. This work uncovered no insurmountable barriers and study progressed to Phase 2.

- Information which would guide the experimental and prototype testing programs needed to complete validation of the use of heat pipe technology for removal of waste heat from Deep Bases.

Following completion of the system integration and optimization study, steps could be taken to validate the use of heat pipe technology for Deep Base waste heat removal applications. The major development steps which would be required are:

1. Testing of a single full-scale heat pipe in a laboratory to check theoretically predicted operation.
2. Testing of multiple, full-scale pipes under conditions which simulate actual operation.
3. Further study of heat pipe mechanical performance, particularly with regard to installation, maintenance, and survivability.
4. Final design and integration of practical Deep Base heat pipe waste removal applications.

The Ballistic Missile Office has expressed a strong desire to have validation of all Deep Base technologies completed by the end of 1986. This objective could be met for heat pipe waste heat removal applications within this time frame, if an aggressive program of study and testing, comprised of the technology development steps described above, were undertaken.

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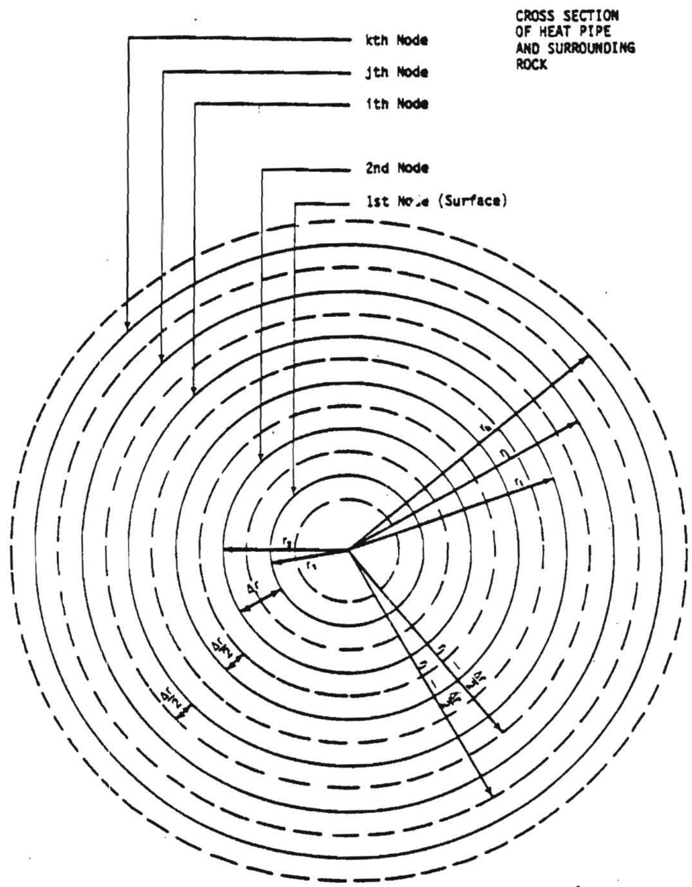
APPENDIX A.1

Derivation of Surface Node Equations for Constant Heat Input

Energy Balance on Surface Node:



SCHEMATIC ILLUSTRATION OF SURFACE AND INTERIOR NODES

(1) $q_{in} = q_{out} + q_{stored}$ (Energy Balance)

(2) $q_{in} = Q''(2\pi aL) = \text{Constant}$

(3) $$q_{out} = 2\pi KL \; \frac{T(1,1) - T(2,1)}{\ln\,[r_1/a]}$$

(4a) $$q_{stored} = \rho CV \frac{dT}{dt}$$

$$\text{(4b)} \quad q_{stored} = \rho C \pi L \, [(a + \Delta r/2)^2 - a^2] \, \frac{T(1,2) - T(1,1)}{\Delta t}$$

Substituting Equations (2), (3), and (4b) into 1 gives,

$$\text{(5)} \quad 2\pi a L Q'' = 2\pi K L \, \frac{T(1,1) - T(2,1)}{\ln (r_1/a)} + \rho C \pi L \, [(a + \Delta r/2)^2 - a^2] \, \frac{T(1,2) - T(1,1)}{\Delta t}$$

Simplifying one obtains,

$$\text{(6)} \quad 2aQ'' = 2K \, \frac{T(1,1) - T(2,1)}{\ln (r_1/a)} + \rho C \, [(a + \Delta r/2)^2 - a^2] \, \frac{T(1,2) - T(1,1)}{\Delta t}$$

Let $B = (a + \Delta r/2)^2 - a^2 = \text{constant}$

Also let $2K\Delta t/(\rho C) = S = \text{constant}$

Now, solving for T(1,2):

$$\text{(7)} \quad 2aQ'' - 2K \, \frac{T(1,1) - T(2,1)}{\ln (r_1/a)} = \rho C B \, \frac{T(1,2) - T(1,1)}{\Delta t}$$

$$\text{(8)} \quad T(1,2) - T(1,1) = [2a\Delta t Q''/(\rho C B)] - (S/B) \, \frac{T(1,1) - T(2,1)}{\ln (r_1/a)}$$

$$\text{(9)} \quad T(1,2) = [2a\Delta t Q''/(\rho C B)] + T(1,1) - (S/B) \, \frac{T(1,1) - T(2,1)}{\ln (r_1/a)}$$

Rearranging Terms:

$$\text{(10)} \quad T(1,2) = T(1,1) \, \{1 - [(S/B)/\ln (r_1/a)]\} + T(2,1) \, [(S/B) / \ln(r_1/a)] + 2a\Delta t Q''/(\rho C B)$$

In order for equation (10) to be valid, each term must be positive, or else T(1,2) may decrease with time -- which is not possible in this case. Let,

(11a) $Term1 = T(1,1) \{1 - [(S/B) / \ln (r_1/a)]\}$

(11b) $Term2 = T(2,1) [(S/B) / \ln (r_1/a)]$

(11c) $Term3 = 2a\Delta tQ''/(\rho CB)$

Now, Equation (10) is as follows:

(12) $T(1,2) = Term1 + Term2 + Term3$

Examination of Term2 and Term3 will show that both of these terms are positive. Therefore, solving for Term1 to insure that it is positive gives:

(13) $Term1 = T(1,1) \{1 - [(S/B) / \ln (r_1/a)]\} \geq 0$

(14) $1 - [(S/B) / \ln (R_1/a)] \geq 0$

(15) $(S/B) / \ln (r_1/a) \leq 1$

Examination of the terms in Equation (15) will show that S is dependent on Δt, and B and r_1 are dependent on Δr. Therefore, Δr and Δt must be chosen such that Equation (15) is satisfied in order for Equation (10) to be valid.

APPENDIX A.2

Derivation of Surface Node Equations for Variable Heat Input

As shown in Equation (C-36), Section II-C, the condenser output can be coupled to the evaporator by the following equation:

$$(1)\quad Q_c = \left[\frac{h_e A_e R_r}{h_e A_e R_e + R_r} \right] \left[T_{in} - T_{p,c} \right] - \frac{C_{HP}\ R_r\ (h_e A_e R_e + 1)}{h_e A_e R_e + R_r} \frac{dT_{p,c}}{dt}$$

Once the heat transfer fluid is chosen, h_e, A_e, R_e, R_r and C_{HP} are constants, therefore, let the following constants be defined:

$$(2)\quad C_1 = \frac{h_e A_e R_r}{h_e A_e R_e + R_r}$$

$$(3)\quad C_2 = C_{HP}\ R_r\ \frac{(h_e A_e R_e + 1)}{h_e A_e R_e + R_r}$$

Assuming perfect contact (no losses) between the heat pipe and the rock surface,

$$(4a)\quad T_{p,c} = T(1,1)$$

$$(4b)\quad \frac{dT_{p,c}}{dt} = \frac{T(1,2) - T(1,1)}{\Delta t}$$

Performing an energy balance on the surface node,

$$(5) \qquad q_{in} = q_{out} + q_{stored}$$

$$(6) \qquad q_{in} = Q_c = C_1 \, [T_{in} - T(1,1)] - C_2 \, \frac{T(1,2) - T(1,1)}{\Delta t}$$

$$(7) \qquad q_{out} = 2\pi KL \, \frac{T(1,1) - T(2,1)}{\ln (r_1/a)}$$

$$(8) \qquad q_{stored} = \rho C \pi L \, ([a + \Sigma r/2]^2 - a^2] \, \frac{T(1,2) - T(1,1)}{\Delta t}$$

Again, let $B = [a + \Delta r/2]^2 - a^2 =$ constant

Substituting Equations (6), (7) and (8) into Equation (5), and solving for T(1,2) yields:

$$(9). \qquad C_1 \, [T_{in} - T(1,1)] - C_2 \, \frac{T(1,2) - T(1,1)}{\Delta t} = 2\pi KL \, \frac{T(1,1) - T(2,1)}{\ln (r_1/a)} + \rho C \pi LB \, \frac{T(1,2) - T(1,1)}{\Delta t}$$

$$(10) \qquad (C_1 \, \Delta t \, T_{in}) - [C_1 \, \Delta t \, T(1,1)] - C_2 \, T(1,2) + C_2 \, T(1,1) = 2\pi KL\Delta t \, \frac{T(1,1)}{\ln (r_1/a)} - 2\pi KL\Delta t \, \frac{T(2,1)}{\ln (r_1/a)} + \rho C \pi LB \, T(1,2) - \rho C \pi LB \, T(1,1)$$

$$(11) \qquad (C_1 \, \Delta t \, T_{in} - [C_1 \, \Delta t \, T(1,1)] + C_2 \, T(1,1) - 2\pi KL\Delta t \, \frac{T(1,1)}{\ln (r_1/a)} + 2\pi KL\Delta t \, \frac{T(2,1)}{\ln (r_2/a)} + \rho C \pi LB \, T(1,1) = C_2 \, T(1,2) + \rho C \pi LB \, T(1,2)$$

$$(12)\quad T(1,2) = \frac{1}{C_2 + \rho C\pi LB} \{(C_1 \Delta t\, T_{in}) + T(1,1)\, [C_2 - C_1 \Delta t -$$

$$\frac{2\pi KL\Delta t}{\ln (r_1/a)} + \rho C\pi LB] + \frac{2\pi KL\Delta t}{\ln (r_1/a)} T(2,1)\}$$

Let $K_1 = \dfrac{1}{C_2 + \rho C\pi LB}$ and $K_2 = \dfrac{2\pi KL\Delta t}{\ln (r_1/a)}$

$$(13)\quad T(1,2) = K_1 \{(C_1 \Delta t\, T_{in}) + T(1,1)\, [(C_2 + \rho C\pi LB) - C_1 \Delta t - K_2] + K_2\, T(2,1)\}$$

$$(14)\quad T(1,2) = (K_1 C_1 \Delta t\, T_{in}) + T(1,1)\, [1 - C_1 K_1 \Delta t - K_1 K_2] + T(2,1)\, K_1 K_2$$

Again, to assure stability in Equation (14), all of the terms must be positive, therefore,

$$(15)\quad T(1,1)\, [1 - C_1 K_1 \Delta t - K_1 K_2] \geq 0$$

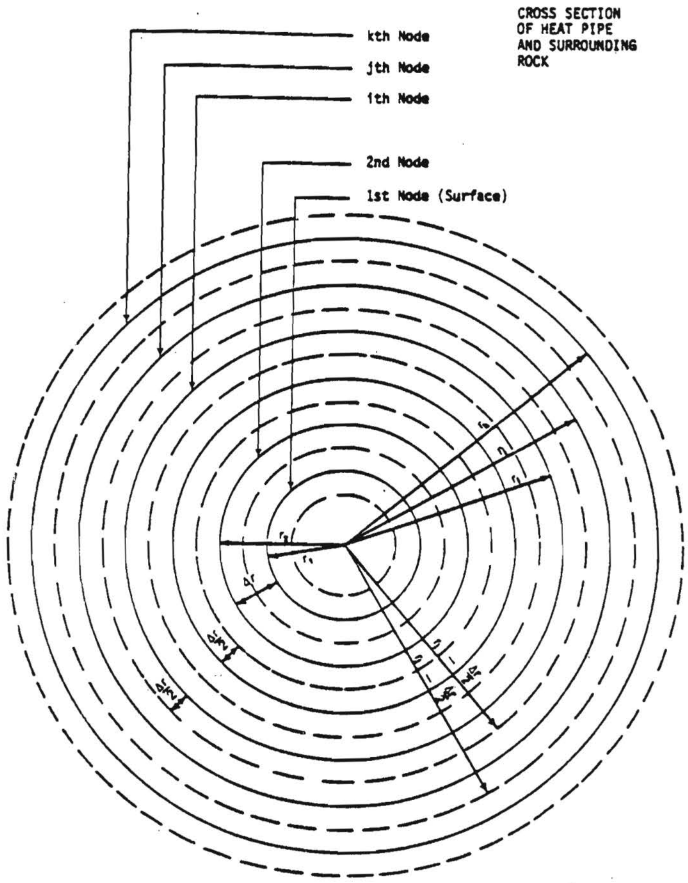
$$(16)\quad C_1 K_1 \Delta t + K_1 K_2 \leq 1$$

As before, Equation (16) will determine which values of Δt and Δr may be chosen.

APPENDIX A.3

Derivation of Interior Node Equations

Energy Balance on an Interior Node:



SCHEMATIC ILLUSTRATION OF SURFACE AND INTERIOR NODES

(1) $(q_{in})_j = (q_{out})_j + (q_{stored})_j$

(2) $q_{in} = 2\pi KL \dfrac{T(i,1) - T(j,1)}{\ln [r_j/r_i]}$

(3) $q_{out} = 2\pi KL \dfrac{T(j,1) - T(k,1)}{\ln [r_k/r_j]}$

(4a) $q_{stored} = \rho CV \dfrac{dT}{dt}$

(4b) $q_{stored} = \rho C\pi L \;\{[r_j + \Delta r/2]^2 - [r_j - \Delta r/2]^2\} \dfrac{T(j,2) - T(j,1)}{\Delta t}$

Substituting Equations (2), (3) and (4b) into Equation (1) will give,

$$(5)\quad 2\pi KL \frac{T(i,1) - T(j,1)}{\ln [r_j / r_i]} = 2\pi KL \frac{T(j,1) - T(k,1)}{\ln [r_k / r_j]}$$

$$+ \rho C\pi L \{[r_j + (\Delta r/2)]^2\} - [r_j - (\Delta r/2)]^2 \quad \frac{T(j,2) - T(j,1)}{\Delta t}$$

Simplifying,

$$(6)\quad 2\alpha\delta t \frac{T(i,1) - T(j,1)}{\ln [r_j/r_i]} = 2\alpha\Delta t \frac{T(j,1) - T(k,1)}{\ln (r_k / r_j)}$$

$$+ \{[r_j + \Delta r/2]^2 - [r_j - \Delta r/2]^2\} [T(j,2) - T(j,1)]$$

To simplify this equation, Let B1 and B2 equal:

$$(7)\quad B_1 = 2\alpha\Delta t / [\{[r_j + (\Delta r/2)]^2 - [r_j - (\Delta r/2)]^2\} \times \ln(r_j/r_i)]$$

$$(8)\quad B_2 = 2\alpha\Delta t / [\{[r_j + (\Delta r/2)]^2 - [r_j - (\Delta r/2)]^2) \times \ln(r_k/r_j)]$$

Substituting Equations (22) and (23) into Equation (21) will yield,

$$(9)\quad B_1 [T(i,1) - T(j,1)] = B_2 [T(j,1) - T(k,1)] + T(j,2) - T(j,1)$$

Solving for T(j,2),

$$(10)\quad B_1 [(T(i,1)] - B_1 [T(j,1)] - B_2 [T(j,1)] + B_2 [T(k,1)] + T(j,1) = T(j,2)$$

Rearranging terms,

$$(11)\quad T(j,2) = T(j,1) [1 - B_1 - B_2] + B_1 T(i,1) + B_2 T(k,1)$$

As before, it is necessary for all terms in Equation (26) to be positive. Since B1, B2, T(i,1) and T(j,1) are all positive, the following inequality must hold:

(12) $T(j,1)\ [1 - B_1 - B_2] \geq 0$

(13) $B_1 + B_2 \leq 1$

Again, it can be seen that B_1 and B_2 are controlled by Δt and Δr; therefore, Δt and Δr must be chosen such that equation (13) holds. Also, Δt and Δr must be chosen such that equation (15) from Appendix A.1 and equation (16) from Appendix A.2 are satisfied.

APPENDIX A.4

Figure A.4-1 on the following page illustrates the theory of superposition. This figure is a side view of the layout of heat pipes for the heat pipe/tunnel header concept showing seven heat pipes imbedded in the rock. The temperature at point A may be determined by the line integral solution, if the heat pipes do not interact thermally. For example, for heat pipe 1:

$$\Theta_1 = T_1 - T_o = \frac{Q'}{4\pi K} \int_{r_1^2/4\alpha t}^{\infty} \frac{e^{-u} du}{u}$$

where:

Θ_1 = Temperature excess

T_1 = Temperature at point A, due to heat pipe 1

T_o = Initial Rock Temperature

Q' = Linear heat flow

K = Rock conductivity

α = Rock thermal diffusivity

t = Time

r_1 = Distance from heat pipe center line to point A

u = Integration variable

The same procedure may be followed -- assuming that each heat pipe is in the rock heat sink by itself -- until a temperature excess at point A for each of the seven heat pipes is found. Then, the temperature excess at point A is found as follows:

$$\Theta_A = \Theta_1 + \Theta_2 + \Theta_3 + \Theta_4 + \Theta_5 + \Theta_6 + \Theta_7$$

and, by definition, the temperature T_A is found by:

$$T_A = \Theta_A + T_o$$

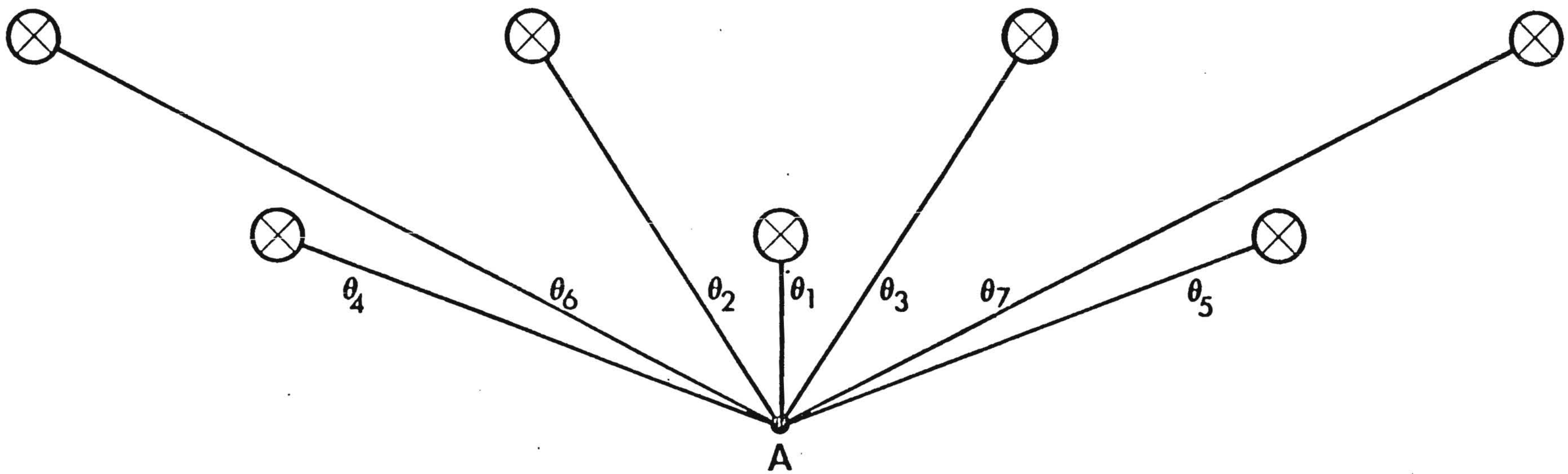


FIGURE A.4-1

SCHEMATIC ILLUSTRATION OF SUPERPOSITION THEORY

In the manner described above, the superposition procedure can be used to determine the two-dimensional temperature profile in the rock for a matrix of heat pipes which interact thermally. However, instead of using a set number of heat pipes to determine the temperature at a given point, all of the heat pipes within a particular distance from the point of interest should be used. For the case at hand, it was found that heat pipes more than 100 feet away from the point being evaluated had negligible influence on the temperature excess.

APPENDIX B
OVERVIEW OF GTSTRUDL

GTSTRUDL is a computer software system for assisting engineers, in structural analysis and design. In essence, this analysis tool, is a sophisticated information processing system capable of supplying accurate and complete technical data for structural design decision making. GTSTRUDL provides the engineer with the ability to specify characteristics of structural problems, perform analyses, reduce and combine results, perform design, and output any part or all of the information stored in the structural problem data base on a selective basis.

GTSTRUDL analytic procedures apply to any combination of framed structures and continuum mechanics problems of arbitrary configuration and composition. Framed structures consist of an assemblage of one-dimensional member elements, which can be represented by properties along a centroidal axis, into any two- or three-dimensional framework. Force boundary conditions on member ends, and force and displacement boundary conditions at support joints, may be specified implicitly by means of structural type and orientation commands, or explicitly for a member or joint. Continuum mechanics problems are treated using the finite element method in which the domain of the problem consists of an assemblage of two- or three-dimensional finite elements of different shapes, connected at a finite number of joints. Over thirty (30) finite elements are available for the solution of plane stress/strain, plate bending, thin shell, and three-dimensional solid problems. GTSTRUDL permits elements (members and finite elements) of different types to be mixed in the same problem solution, whether they have the same or different number of degrees-of-freedom per joint. This characteristic is useful when solving problems such as plates with edge beams, building structures with floor slabs, structures with shear panes, and stiffened shells.

Properties of member elements may be specified by providing section properties of prismatic or variable section members, naming a section from a pre-established table of properties (such as "W14X237"), or specifying flexibility or stiffness matrices for special member elements. Additional conditions for members may be specified such as joint size effects, member end eccentricities from joint centers, location of shear center relative to a member's centroidal axis, etc. Finite element properties may be specified by element type, and either name and thickness, or rigidity matrix (e.g., for anisotropic or orthotropic finite element material properties). Elastic constants may be specified for members and elements.

External influences resulting from applied forces, temperature, initial strain (fabrication error), or specified joint displacements (support movement) may be considered to act separately or in any combination as independent loading conditions. These applied loads may act on members, elements, and/or joints and may have any arbitrary orientation. Loading combinations (dependent loading conditions) may be defined as consisting of any linear combination of independent and other dependent loading conditions.

GTSTRUDL analysis procedures perform linear small displacement static and dynamic analysis of structures composed of any combination of member and finite elements with the same or variable number of degrees-of-freedom per joint.

GTSTRUDL design procedures include steel design and code checking for member elements by the 1969 and 1978 AISC (American Institute of Steel Construction) Specifications for general steel structures, by the 1971 ASCE (American Society of Civil Engineers) Manual No. 52, "Guide for Design of Steel Transmission Towers" for steel transmission tower design, and by the 1980 API (American Petroleum Institute) Recommended Practice for Planning, Designing, and Constructing Fixed Offshore Platforms, for the design of steel tubular members.

The GTSTRUDL user is provided the ability to exercise complete control over a variety of design constraint conditions, similarity specifications, and parameter values. In this manner, iterative design may be performed while carefully controlling the economic and engineering feasibility of the design solution.

GTSTRUDL graphics facilities operate on the line printer (for very low cost plotting), on the CALCOMP plotter (for refined and large off-line plots), and on the TEXTRONIX graphics terminal (for powerful and refined interactive graphics). Extensive graphics facilities are available including the display of general two- and three-dimensional structure geometry and topography, structure deformed and mode shapes, and member force, moment, and envelope diagrams. Powerful special graphics features include 2- and 3-D image rotation, image magnification through windowing, scanning over a structure through panning, extensive annotation and labeling, structure data base display, hidden and boundary line removal, finite element shrink plots, dotted line graphics, split screen plotting, special plot save and restore, as well as many others.

Output may be requested by the user in a variety of formats and in any quantity desired. Output may include input data, joint displacements, support reactions, member end forces and distortions, member force, stress and envelope diagrams at any number of points along a member, element stresses and strains, statics check results, etc. Output may be ordered by loading condition, member, element, or joint, and may be requested for one or more combinations of joints, members, elements, and loading conditions (independent and/or dependent).

One of the most important and beneficial features of GTSTRUDL is its structural data base management facilities. In particular, by appropriate use of the SAVE, RESTORE, ADDITIONS, CHANGES, DELETIONS, ACTIVE, INACTIVE, LOAD, LIST, and other commands, any number of operations may be performed on the structural problem data base input by the user and generated by the system including saving information, modifications or deletion of information, adding new information, etc. For example, by using the SAVE command, the engineer requests GTSTRUDL to take a 'picture' of the current state of a problem solution and save it on a mass storage device for future use. The engineer may then spend as much time as necessary to review and evaluate the latest results of the problem solution, affording the ability to decide upon modifications that need further investigation. After this evaluation, the engineer may RESTORE the problem, specify additions, deletions, or any other changes needed to the problem, and continue the problem solution, including graphics, without having to regenerate the information previously SAVE'd. GTSTRUDL will again execute, referencing the information stored on the computer at the time the last SAVE command was issued, and in addition, consider all new problem constraints and changes made by the engineer.

Additional data base management features include the ability to perform multiple analyses based on the same or changing structural conditions (such as different boundary conditions, and/or different loading conditions, and/or different member and joint topography, etc.) in the same or successive computer runs, the ability to combine analysis results generated in one or more structural analyses involving the same or different structural conditions, the ability to communicate to GTSTRUDL in any desired units, and the ability to change input and output units at any time during a problem solution.

The structural engineer is not required to have any prior knowledge of computers, computer operation, or computer programming in order to analyze and design simple or complex structures. Instead, by using GTSTRUDL, the engineer simply communicates the characteristics of the problem, and procedures to be applied to its solution, by using an English-like Problem-Oriented-Language (POL). The POL is computer independent, easily used and understandable to an engineer, and reflects the terminology a structural engineer would normally use when discussing a problem solution with his colleagues. The POL of GTSTRUDL permits the engineer to dictate his particular problem-solving needs to the computer, rather than having to conform to arbitrary computer program requirements.

GTSTRUDL can be executed in either batch or interactive modes, as well as any combination of batch and interactive modes. GTSTRUDL is the only STRUDL which is available throughout the world for both CDC CYBER large mainframe computers and DEC VAX supermini computers.

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